S. A. E. JOURNAL

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CONTENTS

General Design and Research

Gum in Gasoline—E. B. Hunn, H. G. M. Fischer and A. J. Blackwood 31
Heavy-Duty-Motorcoach Brake-Design—George A. Green 41
Six-Wheel Motor-Vehicle Development History—E. W. Templin 54
Discussion of Six-Wheel Vehicle Papers 59
Friction-Coefficient Research—Louis Illmer 67
Beryllium as a Possible Structural Material—Dr. C. B. Sawyer 98
Automotive Research 99
Standardization Progress 100
News of Section Meetings:
Buffalo 13
Canadian 13
Chicago 115
Cleveland 108
Dayton 116
Detroit 12
Indiana 111
Milwaukee 110
New England 109
Southern California 113
Notes and Reviews 122

Production Engineering

A Production Machine-Tool of Elastic Design—J. B. Armitage 87
Production Engineering 103

Transportation Engineering

Fuel Specifications as Viewed by the Fleet Operator—Adrian Hughes, Jr. 49
Maintenance of Air-Cooled Engines in Commercial Operation—Edward Hubbard and L. S. Hobbs 91
Transportation Engineering 105
News of Section Meetings:
Pittsburgh 111

Aeronautic Engineering

Wings—A Coordinated System of Basic Design—Ralph Upson 15
Demands of Passengers Challenge Engine-Builder's Ability—Robert Insley 38
Miami Aeronautic Meeting, Jan. 14 3
Aeronautic Engineering 102
News of Section Meetings:
Metropolitan 10
Pennsylvania 11

Motorboat Engineering

Motorboat Fire Prevention—H. G. Kamrath 95
Motorboat Meeting, Jan. 22 and 23 3

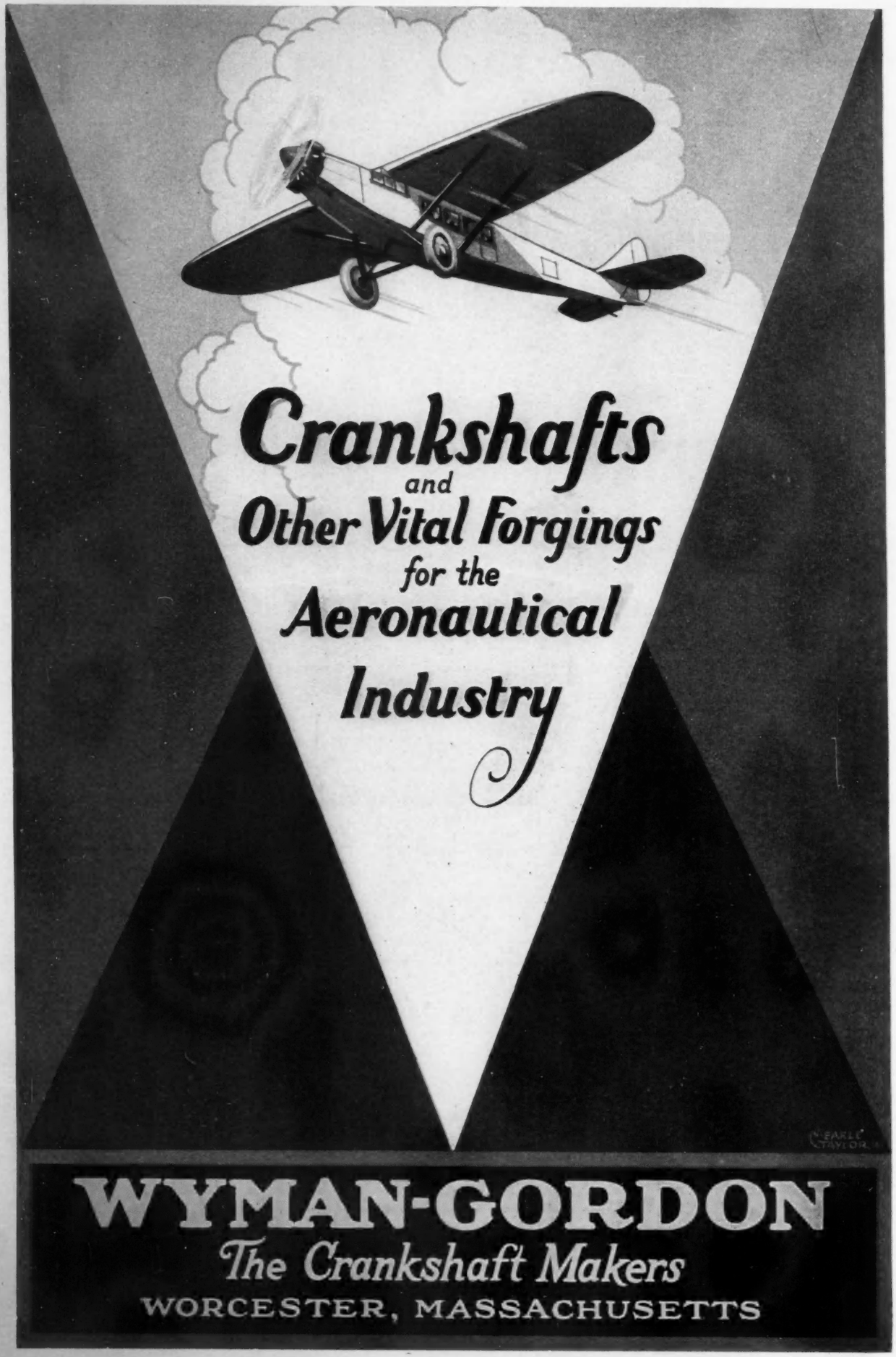
Miscellaneous

Section Membership Values and Opportunities—J. A. C. Warner 96

News of the Society

Events Scheduled for Annual Meeting 1
Miami Aeronautic Meeting, Jan. 14 3
Motorboat Meeting, Jan. 22 and 23 3
Annual Meeting Program 4
Votes on Summer Meeting Location 4
Election of Officers for 1930 5
Council Considers Section Budgets 6
Chronicle and Comment 14
News of Section Meetings 10
Obituaries 117
Personal Notes of the Members 118
Applicants Qualified 120
Applicants for Membership 121

The purpose of meetings of the Society is largely to provide a forum for the presentation of straightforward and frank discussion. Discussion of this kind is encouraged. However, owing to the nature of the Society as an organization, it cannot be responsible for statements or opinions advanced in papers or in discussions at its meetings. The Constitution of the Society has long contained a provision to this effect.


Crankshafts
and
Other Vital Forgings
for the
Aeronautical
Industry
WYMAN-GORDON
The Crankshaft Makers
WORCESTER, MASSACHUSETTS

S. A. E. JOURNAL

Vol. XXVI January, 1930 No. 1

Events Scheduled for Annual Meeting

Important and Timely Topics Arranged for the Many Sessions by the Meetings Committee and the Professional Activities

A "ONCE-OVER" of the 1930 Annual Meeting program on p. 4 of this issue probably will convey the impression of a comprehensive schedule of subjects of general interest to the automotive engineer; more leisurely study of the technical subjects proposed for discussion during the five days of the meeting should produce increased interest and lively anticipation; but thorough consideration of the technical data to be offered and the timely subjects to be discussed will bring the conviction that you must be there.

The Meetings Committee, in cooperation with the various Professional Activities Committees of the Society, has arranged a program that includes subjects representative of the various interests of Society members, the treatment of which is certain to provoke animated and prolonged discussion.

Interesting Sessions on Monday

Following registration in the forenoon, the first day of the meeting, Monday, Jan. 20, will be taken up with the Standards Session at 2 p. m.; a brief business meeting at 8 p. m., and the Production Session, at 8.15 p. m. At the latter session, sponsored by the Production Activity, Dean Dexter S. Kimball, of Sibley College, Cornell University, will discuss The Economics of Production. His views on this vital topic are certain to develop new trends of thought and prove very worth while.

Research Sessions

The Research Committee is sponsoring two sessions: the Cooperative Research Session and the Detonation Session. Bureau of Standards men, whose activities in the field of research are always productive of unusual interest, will be the authors of three papers to be presented at the Cooperative Research Session on the morning of Tuesday, Jan. 21.

At the Detonation Session, scheduled for the afternoon, data on fuel research will be presented in a paper by prominent British engineers, namely, C. H. Sprake, C. H. Barton and R. Stansfield. The second paper to be given at this session is being prepared under the joint authorship of J. M. Campbell, W. G. Lovell, and T. A. Boyd, of the General Motors Research Laboratories.

Aeronautical Subjects

The Aeronautic Session on Tuesday evening, sponsored by the Aircraft and Aircraft-Engine Activities, will consist of two papers and promises to be exceptionally interesting. The authors of these papers, Ralph Upson and Commander E. E. Wilson, will discuss, respectively, The Metalclad Airship and The Importance of Dependability in Aircraft Engines.

Four Passenger-Car Sessions

Four sessions are being sponsored by the Passenger-Car Activity; namely, the Mixture-Distribution Session on Wednesday morning, Jan. 22; the Engine Session on Wednesday afternoon; the Riding-Comfort Session on the morning of the last day of the meeting, Friday, Jan. 24; and the Front-Wheel-Drive Session, scheduled for Friday afternoon.

Considerable interest has been shown in the Mixture Distribution Session, at which papers are to be delivered by Alex Taub, of the Chevrolet Motor Co., and Carl H. Kindl, of the Delco Products Corp., and a great deal of valuable discussion is anticipated.

Following the Mixture-Distribution Session, the Engine Session will be held Wednesday afternoon. The principal papers will be presented by E. H. Shepard, on Downdraft Carburetion, and Louis Illmer, on Friction-Coefficient Research. Prepared discussion of Mr. Illmer's paper, which comprises the results of several years' study of experimentation, will be submitted by a representative of the Chrysler Corp. Mr. Illmer's paper is published in this issue, beginning on page 67.

The Riding-Comfort Session, previously scheduled to be held simultaneously with the Engine Session, has been transferred to Friday morning, Jan. 24. Three papers of outstanding interest and engineering value are to be presented at this session, as follows: Riding-Qualities, by Dr. Fred A. Moss, of the George Washington University; Elimination of Chassis Vibration, by Earl H. Smith, of the Olds Motor Works; and Shock-Absorbers, by Prof. J. M. Nickelsen, of the University of Michigan. Prepared discussion of the last-named paper will be submitted by N. F. Hadley, of the Chrysler Corp. All these men are recognized authorities on the subject.

The fourth session sponsored by the Passenger-Car Activity is to be devoted to front-wheel drives and will start at 2 o'clock, Friday afternoon. William Muller, of the New Era Motors, and J. Ledwinka, of the Edward G. Budd Mfg. Co., will speak on what is perhaps the leading question in the minds of the foremost representatives of the automotive industry today. It may be confidently predicted, in view of the timeliness of the subject and the intense interest manifested in front-wheel drives, that the presentation of Mr. Muller's and Mr. Ledwinka's material will be followed by a popular discussion which will bring out the varying opinions of prominent engineers and may well serve as a basis for determining future engineering development along this line.

Production Events

Paralleling the Engine Session on the afternoon of Wednesday, Jan. 22, will be the Foundry Economics and Production Session, sponsored by the Passenger-Car and Production Activities. Donald Campbell will speak on The Relation of Foundry Practice to Engineering, and A. K. Brumbaugh will discuss Tungsten Carbide Steel in Production.

Alfred Reeves Will Speak

Possibly the most popular and certainly one of the most important sessions will be held at 8 o'clock Wednesday evening, Jan. 22. The Meetings Committee is the sponsor of this, the General Session, which will be devoted to an address by Alfred Reeves, gen-

Meetings Calendar

National Meetings of the Society

Annual Dinner—Jan. 9, 1930
Hotel Pennsylvania, New York City

Miami, Fla., Aeronautic—Jan. 14, 1930
Columbus Hotel

Annual Meeting—Jan. 20 to 24, 1930
Book-Cadillac Hotel, Detroit

Motorboat Meeting—Jan. 22 and 23
Dinner at Hotel Roosevelt, New York City, Jan. 23
Technical Sessions at Westinghouse Lighting Institute

St. Louis Aeronautic—Feb. 18 to 20, 1930
Hotel Jefferson

Detroit Aeronautic—April 8 and 9, 1930
Book-Cadillac Hotel

January Section Meetings

Buffalo Section—Jan. 7
Hotel Statler

Canadian Section—Jan. 17
Royal York Hotel
Speakers—R. H. Combs, Chairman of Section; President, Prest-O-Lite Storage Battery Co., Ltd.

Chicago Section—Jan. 28 or 29
Automobile Show Meeting
Speakers from various automobile companies
Advantages of Supercharging Internal-Combustion Engines—A. C. Mecklenburg, Vice-President and Chief Engineer, Murdock Pump Co., of Indianapolis

Detroit Section
No January Meeting

Milwaukee Section
No January Meeting

Metropolitan Section—Jan. 6
Hotel Commodore, East Ballroom; Dinner at 6.30 P. M.
Automobile Show Meeting
What 1930 Reveals in Engines, Chassis and Bodies

Metropolitan Section—Jan. 22 and 23
Hotel Roosevelt; Dinner at 6.30 P. M., Jan. 23.
Motorboat Meeting

Cleveland Section
No January Meeting

New England Section—Jan. 3
Kenmore Hotel, Crystal Room; Dinner, 6.30 P. M.
Sheet-Steel Stamping of the All-Steel Body—George L. Kelley, Edward G. Budd Mfg. Co.

Northern California Section—Jan. 9
Engineers Club, San Francisco; Dinner, 6.30 P. M.
Carbureters, Governors, Valves and Bearings—Ten-Minute Talks

Southern California Section—Jan. 3
Brakes—Speakers, C. D. Stewart, District Engineer, Westinghouse Traction Brake Co., and Charles Horn, Factory Representative, Booster Brake Co.

Northwest Section—Jan. 11
Seattle, Wash.
Principles of Flight—Walter R. Jones, Project Engineer, Boeing Airplane Co.

Pennsylvania Section
No January Meeting

Pittsburgh Section—Jan. 14
William Penn Hotel
Lubrication

Wichita Section—Jan. 16
Hotel Lassen; Dinner 6.30 P. M.
Paper by C. K. Francis, Ph.D., Chief Technologist, Skelly Oil Co., and Technical Editor, *Oil and Gas Journal.*

eral manager of the National Automobile Chamber of Commerce, on the very timely subject, What 1930 Offers to the Motor Industry. There is no need to enlarge on the desirability of every automotive engineer's "listening in" on this talk; announcement of the subject and author should be sufficient.

Diesel Engines

The fourth day of the meeting, Thursday, Jan. 23, will open at 9.30 a. m. with the Diesel-Engine Session, sponsored by the corresponding Activity. Comprehensive papers have been prepared for presentation by O. D. Treiber, of the Treiber Diesel Engine Corp., and J. E. Wild, of the Robert Bosch Magneto Co., Inc. Titles of the papers are given in the program on page 5.

Transportation and Body Conferences

Thursday afternoon two sessions will be held simultaneously. They are the Transportation Conference and the Body Conference. The former is sponsored by the two Activities directly concerned with this phase of engineering; namely, the Transportation and Maintenance Activity and the Motor-Truck and Motorcoach Activity. The entire conference will be taken up with only one paper, The Trend of Weight and Size in the Development of Motor-Trucks and Motorcoaches, by Pierre Schon, of the General Motors Truck Co. The delivery of Mr. Schon's views on this topic will doubtless be followed by a general voicing of opinions, resulting in an extended discussion.

Body Papers and Dinner

The Body Conference, on the afternoon of Thursday, Jan. 23, has been planned by the Passenger-Car-Body Activity and will consist of two vitally interesting papers on the subjects of Sheet Metal and Its Fabrication for Automobile Bodies, and Body Mounting, to be presented respectively by Dr. George L. Kelley, of the Edward G. Budd Mfg. Co., and Gordon England, of the Holbrook Co.

The Detroit Section Body Division has sponsored and arranged a dinner and session starting at 6.30 p. m. on Thursday. This is sufficient indication to those familiar with similar events staged by the Division that the program will be unusually attractive. The speaker at the dinner will be David Lawrence, publisher and editor of *The United States Daily*, who has chosen as the subject of his address, The Automobile in Perspective.

A Session on Inventions

The Inventions Session, sponsored by the Meetings Committee and scheduled for the morning of the last day of the meeting, Friday, Jan. 24, will provide one of the most novel features of the 1930 Annual Meeting. The work of the New Devices Committee of the General Motors Corp. will be outlined in a paper by W. J. Davidson, and prepared discussion will be submitted by a representative of the Chrysler Corp. The proposal of ingenious new devices and the question of their practicability are always fascinating; therefore the Inventions Session may be depended upon to attract an unusually large attendance and prove one of the most interesting sessions of the meeting.

Those who read the Annual Meeting supplement enclosed in the December issue of THE JOURNAL will note that several changes have been made in the sequence of sessions. No serious conflict of interests is anticipated, however, as a result of the simultaneous sessions that have been scheduled.

Short business meetings will convene at certain technical sessions, as shown in the program herewith, for the purpose of electing Nominating Committees for the respective Professional Activities. Each Nominating Committee so chosen will, in due course, designate a nominee for the Vice-Presidency representing the Activity, in accordance with By-Law 26.

Miami Aeronautic Meeting, Jan. 14

PROGRAMS were issued just before Christmas for the National Aeronautic Meeting to be held in Miami, Fla., on Jan. 14. This is to be a one-session meeting, held on Tuesday evening at the Columbus Hotel. Lieut. Carl B. Harper, U.S.N., is to preside as Chairman. Three addresses are to be given.

The important part that equipment and operation methods play in the financial success of an international airline is to be dealt with by J. M. Eaton, general traffic manager of Pan-American Airways, Inc.

Some of the problems of international commercial aviation will be discussed by Clarence M. Young, Assistant Secretary of Commerce for Aeronautics.

The probable effects of international communication and travel by airlines is the topic on which William B. Stout, president of the Stout Air Services, Inc., will give one of his characteristically vivid and interesting talks.

Members who are in the South, or who are planning to sojourn in Florida at the time, should not forget or overlook this meeting on Jan. 14, as it is sure to be of interest and will give an insight into travel and communication developments that no one questions will have a most important effect on future commercial and political relations with our Central and South American neighbor nations.

Motorboat Meeting, Jan. 22 and 23

Two Technical Sessions and a Dinner To Feature Motorboat Show Week in New York City

AN INTERESTING and comprehensive program has been arranged for the National Motorboat Meeting. This meeting, which is sponsored jointly by the National Association of Engine & Boat Manufacturers and the Society, will be held in New York City on Jan. 22 and 23; that is, Wednesday and Thursday of Motorboat Show week. A technical session will be held on each of the two afternoons at the Westinghouse Lighting Institute on the seventh floor of Grand Central Palace, where the show will be held, and the final event of the meeting will be a dinner at the Hotel Roosevelt, at Madison Avenue and 45th Street, on Thursday evening. The Metropolitan Section of the Society will act as host for the Motorboat Meeting.

At the technical session on Wednesday afternoon the following papers will be presented:

Recent Diesel-Engine Developments, by O. D. Treiber, of the Treiber Diesel Engine Corp.

Boat Standardization from the Cruiser Builder's Point of View, by H. R. Sutphen, of the Submarine Boat Corp.

Boat Lighting and the Need for Standardization, by R. E. Carlson, of the Edison Lamp Works, General Electric Co.

Boat Lighting Recommendations, by Mr. Meeker, of Smith & Meeker.

At the technical session on Thursday afternoon additional papers will be presented as follows:

Safety on Boats, by H. E. Newall, of the National Board of Fire Underwriters.

Safety on Boats, by Jay W. Smith, of the Chris-Craft Marine Motor Co.

Outboard Motors, by Jacob Dunnell, of Ludington Aircraft, Inc.

Racing Motion Pictures.

Mr. Sutphen will act as toastmaster at the Motorboat Dinner at the Roosevelt on Thursday evening. An interesting address by a prominent speaker assures an enjoyable evening. The price of the tickets for the dinner will be $5 each. Requests for reservations, accompanied by remittances, should reach the Society's office, 29 West 39th Street, New York City, as soon as possible. No refunds will be made after Jan. 20.

Votes on Summer Meeting Location

CHOICE of three locations for the 1930 Summer Meeting was submitted late last month to members of the Society by mail, with a return postcard inclosed for indicating first, second and third choice.

The Council and the Meetings Committee, after considering various possible locations, decided to submit to vote the Banff Springs Hotel, at Banff, Canada; Saranac Inn, N. Y., the place of the 1929 Summer Meeting; and French Lick Springs Hotel, at French Lick, Ind.

This year's meeting will mark the 25th anniversary of the founding of the Society, and plans for suitably celebrating the quarter century of growth and accomplishment of the organization are in the making.

Many considerations enter into the selecting of a suitable place. Accommodations for 1000 members and their ladies and guests must be available. Scenic and climatic conditions are important, as are also the facilities for outdoor sports and indoor social events. The adequacy of dining facilities and

Annual Meeting Program

Monday, Jan. 20

10:00 A. M.—REGISTRATION

2:00 P. M.—STANDARDS SESSION

8:00 P. M.—ANNUAL BUSINESS MEETING

8:15 P. M.—PRODUCTION SESSION

(Sponsored by the Production Activity)

The Economics of Production

Dean Dexter S. Kimball, Sibley College, Cornell University

The Session will convene as a stated business session of the Production Activity to name its Activity Nominating Committee to designate a nominee for Vice-President of the Activity for 1931. (*By-Law* 26)

Tuesday, Jan. 21

9:30 A. M.—COOPERATIVE RESEARCH SESSION

(Sponsored by the Research Committee)

The Properties of Gasolines with Reference to Vapor Lock
O. C. Bridgeman, Bureau of Standards

Influence of Atmospheric Conditions on Knock Testing
D. B. Brooks, Bureau of Standards

Effect of Design on Engine Acceleration
D. B. Brooks and C. S. Bruce, Bureau of Standards

2:00 P. M.—DETONATION SESSION

(Sponsored by the Research Committee)

Detonation Characteristics of Some of the Fuels Suggested as Standards of Anti-Knock Quality

J. M. Campbell, W. G. Lovell and T. A. Boyd, General Motors Research Laboratories

Cooperative Detonation Research

C. H. Sprake, Anglo-American Oil Co.; C. H. Barton, Asiatic Petroleum Co., and R. Stansfield, Anglo-Persian Oil Co.

8:00 P. M.—AERONAUTIC SESSION

(Sponsored by the Aircraft and Aircraft-Engine Activities)

The Metalclad Airship—Past Adventures and Future Prospects
Ralph Upson

The Importance of Dependability in Aircraft Engines
Commander E. E. Wilson

Wednesday, Jan. 22

9:30 A. M.—MIXTURE-DISTRIBUTION SESSION

(Sponsored by the Passenger-Car Activity)

Mixture Distribution
Alex Taub, Chevrolet Motor Co.

Cold Carburetion
Carl H. Kindl, Delco-Remy Corp.

2:00 P. M.—ENGINE SESSION

(Sponsored by the Passenger-Car Activity)

Friction-Coefficient Research
Louis Illmer, Brewer-Titchener Corp.

Discussion: Application of the Data
By a representative of the Chrysler Corp.

Downdraft Carburetion
E. H. Shepard, Holley Carburetor Co.

2:00 P. M.—FOUNDRY ECONOMICS AND PRODUCTION SESSION

(Sponsored by Passenger-Car Activity and Production Activity)

Relation of Foundry Practice to Engineering
Donald Campbell, Campbell, Wyant & Cannon

Tungsten-Carbide Steel in Production
A. K. Brumbaugh, White Motor Co.

those for the technical sessions has to be considered. Airport facilities have now become an additional factor to be taken into account. Dates that are available should fall in the latter part of May or in June. Location with respect to distribution or concentration of membership of the Society needs to be borne in mind so that the larger number of attendants at the Annual and the Semi-Annual meetings will not have to spend too much time traveling to and from the meeting places.

Besides the three places submitted to vote, the Council and Meetings Committee considered the Mount Washington Hotel, at Bretton Woods, N. H.; the New Ocean House, at Swampscott, Mass.; the Wardman Park Hotel, at the City of Washington; the Arlington Hotel, at Hot Springs, Ark.; the Grand Hotel on Mackinac Island; the Utah Hotel, at Salt Lake City; and the Chateau Frontenac, at Quebec, the place of the 1928 Summer Meeting.

Election of Officers for 1930

A. L. BEALL, A. C. Bergmann and A. C. Woodbury, named by President Strickland as tellers of election of officers for 1930, have announced the election of those Members nominated by the Annual Nominating Committee of the Society and the Nominating Committees of the eight Professional Activities officially recognized by the Council.

All of these nominees were with very few exceptions elected unanimously. Eleven hundred and fifty-nine ballots were cast, 65 of these being invalid for various reasons. The report of the tellers was as follows:

For President (to serve for one year)

Edward P. Warner	1156
Chance M. Vought	1
O. C. Berry	1

For Vice-Presidents (to serve for one year)

Representing Aircraft Engineering
Chance M. Vought 1158

Representing Aircraft-Engine Engineering
Bruce G. Leighton 1158

Representing Diesel-Engine Engineering
O. D. Treiber 1159

Representing Motor-Truck and Motorcoach Engineering
Arthur J. Scaife 1159

Representing Passenger-Car Engineering
George L. McCain 1159

Representing Passenger-Car Body Engineering
William N. Davis 1159

(*Concluded on p. 7*)

Book-Cadillac Hotel, Detroit—Jan. 20 to 24

8:00 P. M.—GENERAL SESSION

(Sponsored by the Meetings Committee)

What 1930 Offers to the Motor Industry
Alfred Reeves, National Automobile Chamber of Commerce

Thursday, Jan. 23

9:30 A. M.—DIESEL-ENGINE SESSION

(Sponsored by the Diesel-Engine Activity)

Light-Weight Diesel Engines
O. D. Treiber, Treiber Diesel Engine Corp.

Combustion Chambers, Fuel-Pumps and Injection Devices for Solid-Injection Oil Engines
J. E. Wild, Robert Bosch Magneto Co., Inc.

The Session will convene as a stated business session of the Diesel-Engine Activity to name its Activity Nominating Committee to designate a nominee for Vice-President of the Activity for 1931. (*By-Law* 26)

2:00 P. M.—TRANSPORTATION CONFERENCE

(Sponsored by the Transportation and Maintenance Activity, and the Motor-Truck and Motorcoach Activity)

Trend of Weight and Size in the Development of Motor-Trucks and Motorcoaches
Pierre Schon, General Motors Truck Co.

The Session will convene as a stated business session of the Transportation and Maintenance and the Motor-Truck and Motorcoach Activities to name their Activity Nominating Committees to designate nominees for Vice-President of the Activities for 1931. (*By-Law* 26)

2:00 P. M.—BODY CONFERENCE

(Sponsored by the Passenger-Car Body Activity)

Body Mounting
Gordon England, Holbrook Co.

Sheet Metal and Its Fabrication for Automobile Bodies
Dr. George L. Kelley, Edward G. Budd Mfg. Co.

The Session will convene as a stated business session of the Passenger-Car Body Activity to name its Activity Nominating Committee to designate a nominee for Vice-President of the Activity for 1931. (*By-Law* 26)

6:30 P. M.—BODY DINNER AND SESSION

(Sponsored by Detroit Section Body Division)

The Automobile in Perspective
David Lawrence, *The United States Daily*

Friday, Jan. 24

9:30 A. M.—RIDING-COMFORT SESSION

(Sponsored by Passenger-Car Activity)

Riding-Qualities
Dr. Fred A. Moss, George Washington University

Elimination of Chassis Vibration
Earl H. Smith, Olds Motor Works

Shock-Absorbers
Prof. J. M. Nickelsen, University of Michigan

Prepared discussion by N. F. Hadley, Chrysler Corp.

9:30 A. M.—INVENTIONS SESSION

(Sponsored by Meetings Committee)

Work of the New Devices Committee of the General Motors Corp.
W. J. Davidson, General Motors Corp.

Prepared discussion by a representative of the Chrysler Corp.

2:00 P. M.—FRONT-WHEEL-DRIVE SESSION

(Sponsored by Passenger-Car Activity)

Front-Wheel Drive
William Muller, New Era Motors

Bodies for Front-Wheel-Drive Cars
J. Ledwinka, Edward G. Budd Mfg. Co.

The Session will convene as a stated business session of the Passenger-Car Activity to name its Activity Nominating Committee to designate a nominee for Vice-President of the Activity for 1931. (*By-Law* 26)

Council Considers Section Budgets

Favors Making Society and Section Dues One Charge of $20 and Passes Resolutions on Section Entertainment Expense

THE MEETING held by the Council in Detroit on Nov. 26 was attended by President Strickland, First Vice-President Warner, Past-President Wall, Second Vice-Presidents Fisher and Stout, and Councilors Fishleigh, Parker, White and Whittington; also W. N. Davis, G. L. McCain, A. J. Scaife and O. D. Treiber, nominated for Vice-Presidencies during 1930.

The Council considered detail budget figures submitted by the various Sections.

Section Entertainment Expense

At the June 5, 1929, Council meeting the following resolution had been passed regarding Section entertainment expense:

In order to clarify the situation with regard to Sections entertainment expense, all previous action of the Council with regard to this matter was, upon passage of resolution to reconsider, rescinded; and upon motion, duly seconded, the following policy was approved:

A reasonable amount of first-class entertainment is recognized as a desirable part of regular Section meetings, and, in order to promote this activity, the Council recommends that $50 for an attendance up to 150; $100 above 150 and up to 350; and $150 for more than 350 attendance at any one meeting, be considered a legitimate Section expense.

It is the judgment of the Council that, in general, the total amount spent for entertainment at any one Section meeting should not exceed the respective amounts specified.

After a long discussion at the Nov. 26 Council meeting, in view of the budget expense figures submitted by some of the Sections, the Council voted that the following statement should be sent to all of the Sections, in conjunction with the foregoing resolution passed by the Council regarding Section entertainment expense at its June 5, 1929, meeting.

In those cases where such entertainment expenditure is desirable for a Section, every effort should be made to provide for the covering of cost out of local income directly connected with the meeting. And the assumption of entertainment expense as a net addition to the budget will be approved only where it is apparent to the Council that the special circumstances of the Section make it impracticable to secure the necessary income locally for that purpose.

The experience of the Council indicates that, where 200 or more attend a meeting, the whole entertainment expense should be covered by the income of that meeting.

Section Budget Forms

After discussion of the advisability of asking the Sections to submit on specially prepared forms, as in former years, estimated budget expenses for a given year, it was felt that there should be such a form. Accordingly, a form was prepared and sent to the Sections, calling for general information as to Sections activities and estimated income and expense figures for the current year, in parallel with actual income and expense figures for the last previous year.

Society Financial Report

A financial statement as of September, 1929, was submitted and approved. This statement showed a net balance of assets over liabilities of $229,764.86, this being $18,706.70 more than the corresponding figure on the same day of 1928. The gross income of the Society for the fiscal year ended Sept. 30, 1929, amounted to $405,138.08, the operating expense being $386,431.38. The income for the month of September was $34,144.03, and the operating expense during the same month $29,-063.50.

Membership Matters

Thirty-eight applications for individual membership in the Society were approved, as well as 6 transfers in grade of membership; and 180 applications for membership, 21 transfers in grade of membership, 15 reapprovals and 1 reinstatement, on which the Council had acted by mail vote, were confirmed. Thirty-eight resignations were accepted and 11 reinstatements to membership were made. Two hundred and thirty-four members were stricken from the rolls of the Society for non-payment of dues accrued Oct. 1, 1928.

Society and Section Dues Proposal

It will be recalled that for some years there has been discussion of the matter of "making Society and Section dues one charge of $20." At the time of the last discussion the feeling was that this charge should be made to Members and Associates residing in the United States, Canada, Mexico and Cuba.

The matter was brought up at the November Council meeting with a view to the Council expressing opinion as to the desirability of introducing the necessary Constitutional amendment at the 1930 Annual Meeting of the Society to be held in Detroit this month.

In connection with the proposed change, it was the view of the Council at its meeting held in June that each Section territory, in the event of Constitutional change as aforesaid, should consist of an area included by a 50-mile radius from the Section headquarters. It was also reported that the Finance Committee felt that making the Society and Section dues one charge of $20 would virtually close the doors to a further general increase in dues for some time to come.

The Council at its June meeting approved as a matter of policy the changes that would be in effect *if* Constitution and By-Laws amendments, both Society and Section, were made as shown by drafts of amendments that were submitted and included in the minutes of the June 5, 1929, Council meeting. These drafts of amendments were as follows:

SOCIETY CONSTITUTION AND BY-LAWS

C 21. The annual dues for membership in each grade and for Student Enrollment shall be as follows:

For Member	$20.00
For Associate	20.00
For Junior	10.00
For Service Member	15.00
For Foreign Member	15.00
For Departmental Member	None
For Each Affiliate Member Representative	20.00

NOTE.—Each Affiliate Member Representative would receive the publications of the Society on the same basis as individual members receive them and his name would be included in alphabetical, geographical and company lists of the Society Roster.

C 21a. The Society shall turn over to each Section of the Society a part of the Society dues, which shall be specified in the By-Laws, paid by each Member and Associate and for each Affiliate Member Representative residing in the territory of each Section as defined by the Council.

B 31a. Upon payment of Society dues by each Member and Associate and for each Affiliate Member Representative, there shall be available for the use of the Section in whose territory the Member, Associate or Affiliate Member Representative resides, an amount of $5 which shall be considered as Section dues collected for the Section by the Society. This sum shall be remitted to the Section Treasurer as and if needed to carry on Section activities of approved types. Additional sums may be granted to Sections in the discretion of the Council.

SECTION CONSTITUTION AND BY-LAWS

SC 6. All members of the Society and Affiliate Member Representatives in good standing residing in Section territory shall be members of a Section. A Society member or an Affiliate Member Representative not residing in Section territory and not already a member of any Section may upon request become a member of a Section.

SC 7. Each member of the Section who is present in person at any meeting shall be entitled to one vote on any question sub-

mitted, except that no one who does not hold voting power in the Society under the Constitution thereof shall vote in the annual election by the Section of a member of the Nominating Committee of the Society, as provided in the Constitution of the latter.

SC 9. The annual dues for Section membership are included in the Society dues, the amount of the dues for each member in Section territory being paid to the Section Treasurer by the Society in accordance with rulings of the Council of the Society. There shall be no initiation fee.

SC 10. A member whose Society dues remain unpaid for three months shall be considered as not in good standing in the Section and shall not be entitled to vote or enjoy any other privilege accorded Section members in good standing until dues in arrears shall have been paid in full.

SB 9. The Society will pay to the Section as Section dues for each Member, Associate or Affiliate Member Representative residing in the Section territory [whose Society dues are paid] an amount of $5. This sum shall be remitted to the Section Treasurer as and if needed to carry on Section activities of approved types. Additional sums may be granted to Sections in the discretion of the Council.

Student Branch Enrollment

It will be noted that in the draft of amendment of C21 given above, mention of Student-Enrollment annual fee has been omitted. The basis of this was that it was thought that the fee for Student Enrollment should not be specified in the Constitution as provision as to this should be sufficiently flexible, the matter being suitably covered in the By-Laws by general provision as to Student Enrollment; the Council fixing the amount of the Student-Enrollment fee. If such a change were made, mention of Student Enrollment in C20 should also be deleted.

The Council also considered the recommendation of the Student-Branch Committee that the second paragraph of C51 of the Society's Constitution be amended to read as follows:

The Council may further, in its discretion, authorize the enrollment, individually or by group, of persons under 30 years of age who, at the time of application, shall be bona-fide students of a recognized institution of engineering or pursuing a course of study in [automobile] *or aeronautical* engineering. Student Enrollment [shall] *may* be for [a term of three years, which shall not be renewable] *the duration of the student's college course plus one year thereafter.*

NOTE.—Proposed omissions are enclosed in brackets. Proposed additions are printed in italics.

It was the sense of the Council that the suggested change as to duration of term of Student Enrollment is proper but that, instead of adding the words "or aeronautical," the word "automobile" should be changed to "automotive" in C51 of the Society's Constitution.

It was also the sense of the Council that the following sentence should be added to C51: "The Council shall determine the amount of the enrollment fee."

Constitutional amendment as aforesaid was not proposed officially at the Summer Meeting of the Society held at Saranac last June, because the proposed amendments in mind with relation to Student Enrollment were closely related to the amendments discussed with regard to Section dues; the latter amendments also not having been proposed officially at the Society meeting at Saranac.

These two groups of amendments were listed for Council consideration for the purpose of securing an understanding to avoid confusion and insufficiently deliberate action, if any, at the Annual Meeting of the Society in Detroit this month.

At the Nov. 26, 1929, meeting it was the view of the Council that, after studied consideration, it favored the various Constitutional amendments drafted for informal study during the last year or so. This merely gives in advance the view of the Council on the various matters. The amendments in mind have no official status at this time, under the provisions of the Society's Constitution, and will have no such status until they shall have been proposed in writing by a voting Member of the Society at an Annual or Semi-Annual Meeting and duly seconded by a voting Member.

The drafts of amendments are printed in THE JOURNAL at this time so that the Members in general can study them before the time of the Business Session of the 1930 Annual Meeting of the Society, on Monday night, Jan. 20.

Election of Officers

(*Concluded from p. 5*)

Representing Production Engineering	
John Younger	1155
E. P. Blanchard	1
Representing Transportation and Maintenance Engineering	
F. C. Horner	1158
J. F. Winchester	1
For Members of the Council (to serve for two years)	
Ralph R. Teetor	1159
Frederick K. Glynn	1158
Arthur W. S. Herrington	1158
For Treasurer (to serve for one year)	
C. B. Whittelsey, Jr.	1158

Tellers of Election
A. L. BEALL
A. C. BERGMANN
A. C. WOODBURY

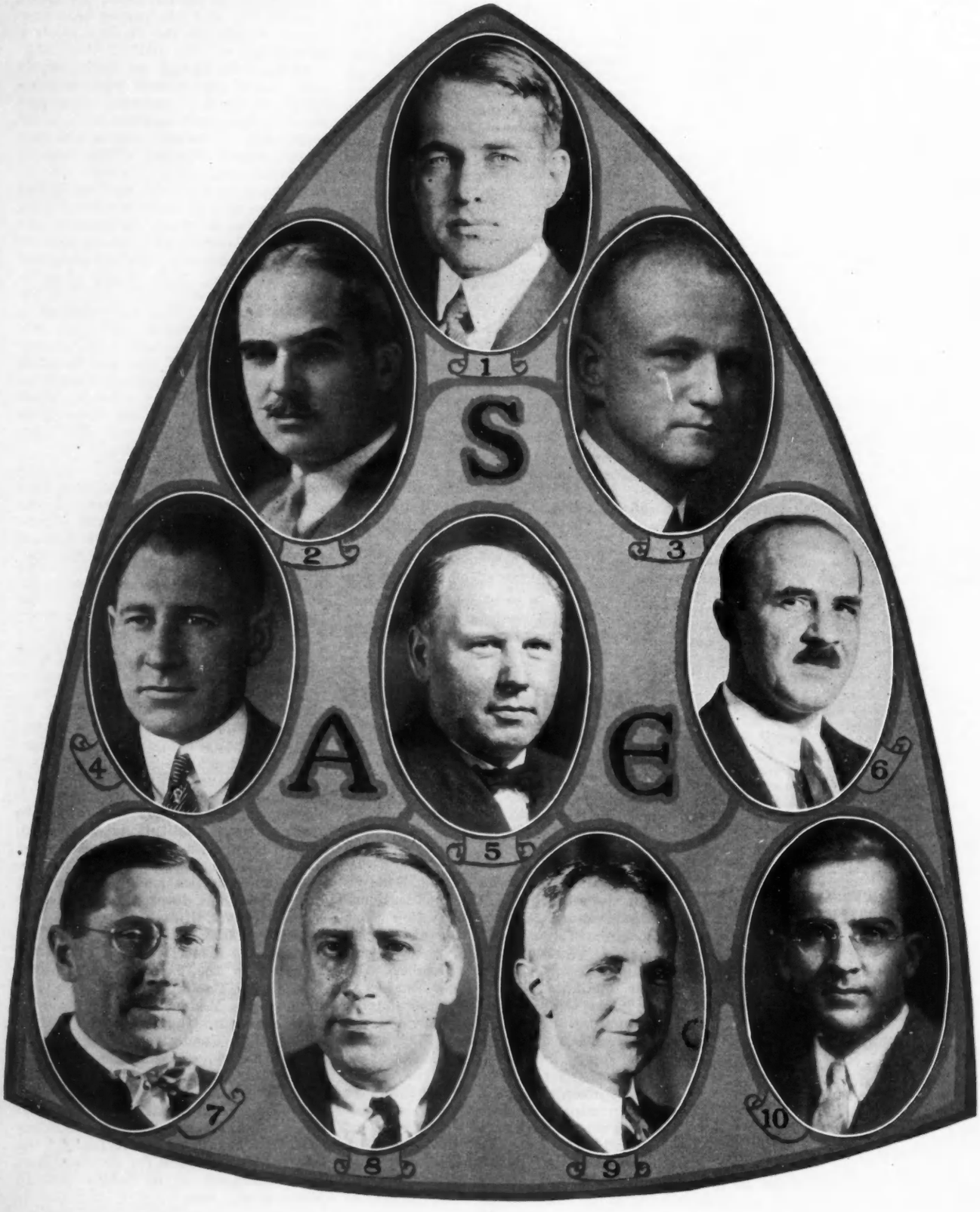
The members listed above as receiving scattering votes were, of course, not on the tickets of the Nominating Committees.

Change in Method of Election

As stated elsewhere in this issue of THE JOURNAL, p. 14, the constitutional provisions of the Society which were in effect for many years have been changed, with the result that, beginning with this year, the Members who are nominated for Society offices or Council memberships are not named by one Nominating Committee. Of course, as heretofore, there may be independent Nominating Committees to select Members to run for the offices.

The change consists in the fact that the nominations for the Vice-Presidencies, which are representative of the Professional Activities officially recognized by the Council, are not made by the Annual Nominating Committee of the Society or by a special Nominating Committee authorized to organize by Constitutional provision, but by Nominating Committees elected at stated Business Sessions of the Professional Activities held each year, at either meetings devoted solely to the respective Activities or meetings one or more sessions of which are sponsored by the respective Activity. However, as in the case of the Annual Nominating Committee of the Society, 20 or more Members entitled to vote may constitute themselves a Special Nominating Committee of a Professional Activity, with the same power as that of the Annual Nominating Committee of its Activity, with the restrictive provision as to time that a special Professional Activity Nominating Committee, if organized, must present the name of its candidate for the respective Vice-Presidency on or before Oct. 15 in the given year.

Officers and Councilors of the



Society Elected to Serve for 1930

OPPOSITE PAGE—PRESIDENT AND VICE-PRESIDENTS OF THE SOCIETY FOR 1930

(1) Edward P. Warner, President; (2) Chance M. Vought, Vice-President Representing Aircraft Engineering; (3) Bruce G. Leighton, Vice-President Representing Aircraft-Engine Engineering; (4) Otis D. Treiber, Vice-President Representing Diesel-Engine Engineering; (5) Arthur J. Scaife, Vice-President Representing Motor-Truck and Motorcoach Engineering; (6) George L. McCain, Vice-President Representing Passenger-Car Engineering; (7) William N. Davis, Vice-President Representing Passenger-Car-Body Engineering; (8) John Younger, Vice-President Representing Production Engineering; (9) Frederick C. Horner, Vice-President Representing Transportation and Maintenance Engineering; (10) Charles B. Whittelsey, Jr., Treasurer

ABOVE—COUNCILORS OF THE SOCIETY FOR 1930

To Serve for Second Year of Office: (1) Walter T. Fishleigh; (2) James A. Moyer; (3) Orrel A. Parker. Elected To Serve during 1930 and 1931; (4) Frederick K. Glynn; (5) Arthur W. Herrington; (6) Ralph R. Teetor. Past Presidents of the Society To Serve as Councilors; (7) William R. Strickland; (8) William G. Wall

Met. Section Considers Deck Landings

Lieutenant Webb Tells About Plane Operation on Carriers and Difficulties of Roof Landings for Transport Planes

HON. EDWARD P. WARNER was introduced as the next President of the Society, when Chairman George A. Round called upon him to preside at the Dec. 19 meeting of the Metropolitan Section. Mr. Warner remarked on the incidental dividends that are being returned to the automotive industry on the necessary expenditures for the War and Navy Departments when he, in turn, introduced Lieut. L. D. Webb as one of those largely responsible for the success of airplane carriers by developing the technique of operation.

Before reading his paper, Lieutenant Webb sketched the development of operations from the first catapult launching of the late Commander Ellison in 1910, when he was a naval lieutenant, and the first landing of the same pilot on the deck of the armored cruiser Pennsylvania in January, 1911, to the present time. Catapults 150 ft. long were placed over the sterns of two armored cruisers before the war; but they were removed when the vessels were needed for convoy service. The British did some rather dangerous "turret jumping" with small single-seaters powered with rotating engines when observation service was needed, but the turret runways interfered with the gun-mountings, and the pilots were obliged either to fly to land or take their chance of being fished out of the water.

The first catapults were operated by compressed air, but gunpowder is generally used now, said Lieutenant Webb, because it renders the operation simpler and independent of the rest of the ship. The catapults are about 60 ft. long and will handle planes having a maximum weight of 6000 lb. Catapulting will give a speed of 55 or 60 m.p.h. to a 5000-lb. plane in 1½ sec. This means an acceleration three times that of gravity, which creates a sensation that is not unpleasant but will not be forgotten. Launching from a catapult is not difficult for the aviator, but to land again on the ship is very difficult, especially if the vessel is a crusier rather than a battleship. Frequently the stern of the boat is swung to "slick" the water.

Airplane Carriers Most Potent Weapons

When a collier was converted into the first carrier, the Langley, it became a pioneer also of electric propulsion. The arresting gear used on this ship consisted of cables pulling weights in vertical channels. Until these weights also were provided with arresting gear, the landing airplane was pulled backward again after it stopped. On the whole, the weights proved satisfactory and are still in use on the Langley. Operating 42 aircraft from the deck of the Langley was said by Lieutenant Webb to be a greater feat than operating 100 from one of the newer carriers.

After the Naval Armament Conference of 1922, two partially completed battle cruisers were converted into the 125,000-ton carriers Lexington and Saratoga. Newspapers have criticised the expenditure of $40,000,000 on these vessels, but Lieutenant Webb believes that half of the cost was due to the fact that they were not originally designed for their final purpose. He said that they are the most potent weapon possessed by any nation. They can deliver an air attack over a range of 1000 miles within 24 hr., as was demonstrated in recent air maneuvers. Each of them carries 80 planes and has facilities for rebuilding 7 at once. They can outsail the Bremen, and one of them covered the distance from Los Angeles to Honolulu in 72 hr. in 1928.

All taking off and landing on the decks of carriers is controlled by an officer on deck. Launchings and landings to the number of 14,000 to 15,000 have been accomplished on the Langley and about 4000 each on the Lexington and Saratoga; and no one has yet been injured in these operations, although a number of pilots have gone over the side without flying speed and been ducked and more or less bruised.

When airplanes are standing on deck they are placed so close together that it is unsafe to pass between the tail of one and the propeller of the next when the propeller is rotating. Wind velocity on the deck caused by the speed of the boat steaming into the wind and augmented by the slipstream of many propellers often reaches 50 m.p.h. Once a heavy bomber started soaring like a kite in this wind and came down on other airplanes. It is necessary to use chocks on both sides of the wheels and to keep each airplane tied down securely until its turn comes for taking off. The safety that has been attained has resulted in part from special training of the pilots and the development of landing-gear and other parts of the airplanes to withstand the unusual stresses encountered.

Stationary Platform Is Unlike Carrier

After laying in this background, Lieutenant Webb read his paper, in which he stated that commercial airplanes can be modified to be suitable for landing on restricted platforms on land and taking off with the aid of catapults. One of the changes necessary in many planes would be better provision for vision from the pilot's cockpit. The problem is more difficult than on shipboard because of lack of the aid given by sailing the carrier into the wind. The heavier wing-loading customary on transport planes also adds to the difficulty, and the great additional responsibility placed upon the pilots of passenger-carrying airplanes would increase to an undesirable extent their "passenger consciousness."

Difficulty is seen also in finding central sites that will not involve much low flying over thickly populated areas. No way is seen to prevent disaster to the occupants of the airplane and to users of the streets from airplanes running over the edge of the platform.

AIRPLANES READY TO TAKE OFF FROM THE LEXINGTON

In naval operation, this has resulted in nothing more serious than a wetting and a few bruises.

Lieutenant Webb's estimate is that a landing area 250 x 1000 ft. would be the minimum size that should be contemplated for the large transport airplanes of the immediate future, and that it will be cheaper to build two such runways at 90 deg. than to mount one on a turntable so that it can be headed into the wind. Over-water approaches would be most likely to be clear and appropriate, he believes, and the availability of platform operation is more probable for air mail than for passengers, especially since post offices in some cities are large enough for the roofs to be used as landing platforms.

Vivid Motion Pictures of Operation

Carrier operation was visualized for the 155 who attended the meeting by three reels of motion pictures. The first two reels showed normal take-offs and landings, including the launching of a whole deck-load of planes in succession and slow-motion pictures of the action of the arresting gear. Lowering the planes on the elevators and moving them to their places below decks was shown, and the control exercised by the control officer was clearly visible. This included signaling to show that the pilot might or might not land, directing the height of the approach and when to cut the spark. Even more interesting was the third reel, composed mostly of slow-motion pictures taken by a cameraman who is ready during all landings and turns his crank whenever it is evident that a bad landing is likely. Most of the landings shown were of pilots who were receiving their training on carrier service, and many of them occurred before the arresting gear of the Langley had reached its present excellence and the landing-gear of the planes was strengthened for the service. Chairman Warner recognized one of the bad landings as one in which he was a passenger, but the fact was not announced early enough to give the engineers a chance to fully appreciate the joke.

Too Much Aviation on the Ground

In the absence of Police Commissioner Whalen of New York City, lack of whose comments on the problem of low flying was regretted by Chairman Warner, A. P. Taliaferro, Jr., of the Aircraft Division of the Department of Commerce, was the first speaker. He quoted a statement that 90 per cent of aviation is on the ground. He believes that the most probable means of reducing the waste of time at the end of flights is by improvement of ground-transportation facilities and the provision of fields that are as central as possible; storage and repair facilities being supplied by outlying fields. Relatively few airports will be developed with public funds, because of the difficulty in securing appropriations. Consciously or unconsciously, real-estate speculation is an important feature of an airport. The amusement feature can be made a source of profit; 5000 to 15,000 people ordinarily visit the Oakland airport in California during a week-end.

Substitutes for Platforms Suggested

Representatives of two manufacturers of amphibian planes expressed the opinion that planes of this type might well be used without the need of landing platforms, if it would be advisable to have such platforms located at the waterside. If such planes are not to be used for long-distance facilities, they can at least be used for shuttle service from the landing-fields of the cross-country airplanes.

A suggestion reported was that a wire mesh, having an area large enough to land any airplane, might be provided, but Lieutenant Webb said that a study of the idea in the Navy had led to its abandonment because of its structural limitations. Replying to a question, he stated an opinion, based on experience on the experimental landing-platform at Hampden Roads and cross-wind landings on carriers, that safe landings could hardly be made in cross winds higher than 18 to 20 m.p.h., even with light airplanes.

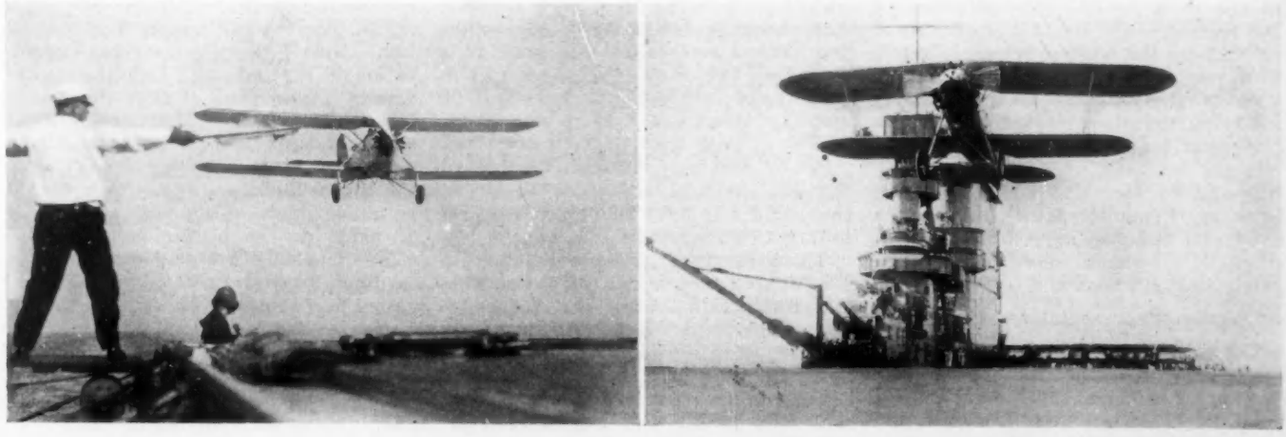
An enthusiastic vote of thanks was given to Lieutenant Webb, and most of those at the meeting stayed to see a final film showing naval aeronautics, made for the purpose of encouraging enlistment in the Navy.

Activities of the Metropolitan Section during January include the Annual Automobile Show Meeting at the Hotel Commodore, Monday, Jan. 6, at which features of 1930 models will be discussed, and the National Motorboat Meeting, Jan. 22 and 23, during the Motorboat Show, with dinner at the Hotel Roosevelt the second night. Among the subjects to be considered at the latter meeting are outboard engines, safety on boats, boat-lighting requirements, boat standardization, Diesel-engine development and racing.

Flying Safety and Geared Aircraft Engines

SEVERAL phases of aeronautic engineering were discussed by scheduled speakers and members in attendance at the meeting of the Pennsylvania Section in Philadelphia on Dec. 19. Thirty members were present at the dinner in the rooms of the Philadelphia Automobile Trade Association, which was followed by entertainment features that included banjo music and some "comedy." The technical session was attended by 65 members.

W. Lawrence LePage, the first speaker, reviewed the elementary theory of aerodynamics, showed that from



LANDING AND TAKING OFF ON THE LEXINGTON

The Bomber at the Left Is Landing as Directed by the Control Officer in the Foreground. The Fighter at the Right Is Taking Off. Both Photographs Were Taken from the Edge of the Deck

this standpoint the airplane is safe in flight, and said that its safety should be judged on this basis rather than by the number of accidents. He compared the present development stage of the airplane with the early days of the automobile and remarked that it is unfair to compare the present accident rate of airplanes with that of motor-vehicles. One reason that the public fears to use an airplane is that they cannot see the air on which it moves and therefore think of it as unsubstantial. If the wheels of an automobile that rest on the ground were invisible, people would be afraid to ride in motor-cars. Mr. LePage's paper was illustrated with 30 lantern slides.

Some experiences with parachute jumping were related by Mr. Ford, of the Irving Air Chute Co.

The development history of geared aircraft engines was reviewed by Section Secretary E. B. Neil, manager of the research department of *Automotive Industries*. The first airplane to fly by power, in 1903, he said, had a geared engine. He told of work done on the problem in and subsequent to the war period, saying that most of the development had been in Europe. His talk was illustrated with 15 lantern slides.

In the discussion on geared engines, H. L. Brownback, of the Brownback Motor Laboratories, told about his early work on and later experience with the use of geared engines and described the design of some of them. He spoke of the troubles encountered in Europe with certain types of European geared engine.

Mr. Randall, of the Naval Aircraft Factory, also participated in the discussion on this subject. Mr. Cochrane, also from the Aircraft Factory, placed emphasis on the point that in almost all cases of accident that have been analyzed the cause had been traced to negligence by some individual; the airplane involved was not really unsafe but may have been improperly inspected or someone had overlooked making some essential repair, replacement or adjustment.

Others who took part were Mr. Mayer, of the J. G. Brill Co., O. D. Treiber, of the Treiber Diesel Engine Corp., E. J. Waring, of the Pyrene Mfg. Co., and A. Gelpke, of the Autocar Co.

Engineering for Sales

H. S. Vance Tells Detroit Section Engineers the Importance of Developing Commercial Instinct

ENGINEERING of the product is the most important part of selling. A proper development of the commercial instinct in engineers will make them the most potent salesmen in their organizations.

This was the theme of a highly interesting address given by H. S. Vance, vice-president in charge of engineering and manufacturing activities of the Studebaker Corp., at the Dec. 9 meeting of the Detroit Section. The dinner at the Book-Cadillac was attended by 208 members, and about 240 were present to hear the address which followed. This was felt to be a good showing in view of the reaction in the industry following the stock-market tail-spin in November, and optimism was injected into the remarks by Chairman L. C. Hill and by Mr. Vance. The Section Greeting Committee, the dinner, the band and the dancing soubrettes contributed the usual good cheer and gaiety to the occasion.

Section Representatives Elected

The first piece of business following the convening of the meeting was the election of Section representatives as follows:

On Nominating Committee of the Society—P. J. Kent, of the Chrysler Corp.; alternate—B. J. Lemon, of the United States Rubber Co.

On Sections Committee of the Society—J. M. Crawford, of the Chevrolet Motor Co.

In his comments leading up to the introduction of the speaker of the evening, Chairman Hill narrated several incidents showing how, by initiative, enterprise, confidence and energy, one sales organization consummated six times as many car sales at the recent Chicago Automobile Salon as any other exhibiting company and sold about 30 automobiles at the New York Automobile Salon by the same tactics, while the nearest competitor sold six. In a period of two weeks the same organization sold in the neighborhood of 50 cars in Detroit, all at prices above $4,000. "This shows," said Mr. Hill, "that if you go out and work hard enough, you will find some people in the world who have enough money left to buy an automobile."

These remarks led appropriately to Mr. Vance's talk on the topic, Commercial Instinct in Automobile Engineering, which might very well have been entitled, Engineering Automobiles to Sell. Pointing out that the automobile dealer, unlike most merchants, has no choice in buying the merchandise he sells, having delegated that important function to the factory organization in exchange for the privilege of selling its product exclusively in his territory, Mr. Vance said that the responsibility of producing a car that will be salable rests largely upon the engineer of the manufacturing company. Success of their work is not determined in their laboratories but in the salesrooms of the dealers. An idea that may be successful from an engineering viewpoint may be an expensive failure from a merchandising viewpoint. The engineer should count his accomplishments in terms of sales and profits. The test that should be applied to engineering ideas, large and small, is, Will this sell? If the answer is No, or is uncertain, constructive engineering will give a negative verdict.

Common sense and information form the basis on which an engineer can decide which of his ideas will sell, asserted the speaker. But common sense is an elusive element in business because it gets snarled up with hope and pride and personal gratification. There is no Paris to which the engineer can go in the fall, as the costumer does, to see on display the models that experience has proved the public will buy in the spring, but the engineer can make many contacts with the buying public in his own organization, through the sales department and by correspondence or personal contact with the company's dealers. It is as important for the engineer to travel the Country obtaining first-hand information on the salability of his product as for the sales manager to travel to obtain information on the adequacy of his sales methods.

The Engineer's Cardinal Function

Mr. Vance then drew a contrast between engineering ideas that increased car salability, such as engine starters, four-wheel brakes, safety glass and shock-absorbers, and those which added little if any sales appeal because they lack laymen interest; for example, a slight change in wheelbase or in piston displacement, an increase in area of braking surface or in bearing area, or an increase of 1 or 2-sec. acceleration. Nearly every engineering organization in the industry, he believes, has wasted a great deal of time and money in making changes that are without sales value. Too much time and money are spent in engineering departments chasing will-o'-the-wisps which should be nailed down to a practical analysis of value at the beginning. Development of present accepted design is the cardinal function of engineers, suggested Mr. Vance, together with the injection of freshness and style into the product from time to time as circumstances require.

Engineers are, for the time being, at least, in the position where the public is without interest so far as anything except constructive engineering is concerned, asserted the speaker. A proper development of the commercial instinct

in engineers will make them the most potent salesmen in their organizations. It is inevitable, he said in conclusion, that from time to time new ideas, even revolutionary ideas, will come into this industry, but those great ideas are rare. Between such unusual events, by using common sense, trying to keep in touch with the public and asking ourselves what it will buy, we have a chance to do our individual organizations and our industry a great service.

Interested in Cold Starting

Canadian Section Hears Oil Technologist Tell of Lubricants Development Problems

OIL companies are cooperating with the automotive engineers in trying to get the best results in the operation of automobiles and are entitled to considerable sympathy in the problems they are facing, smilingly remarked W. A. P. Schorman, chief technologist of the British-American Oil Co. of Canada, at the Dec. 19 meeting of the Canadian Section in Toronto. His address on lubricants and modern methods of their production and testing elicited much interesting discussion.

Chairman R. H. Combs, who presided, was elected as Canadian representative on the Society's Nominating Committee and Vice-Chairman A. S. McArthur was elected as representative of the Section on the Sections Committee.

Viscosity Uniform at Transition Point

The oil companies of the United States spent approximately $9,000,000 in 1929, remarked Mr. Schorman, for new equipment in connection merely with the extraction of waxes from lubricating oils, to reduce the present pour-test and at the same time keep shear down to normal conditions. He then discussed the properties of the various crude oils and the serious problem of dewaxing. With a series of lantern slides he showed how the different oils behave under varying conditions of friction and temperature, but he maintained that the viscosity of the different oils, no matter of what grade, is exactly the same at the point of transition. Recent experiments, he said, have proved this and he showed a graph illustrating how three different oils reached the transition point at 217, 246 and 260 deg. fahr., but had exactly the same viscosity.

Wide Variation in Starting Quality

Of particular interest to Canada and the northern sections of the United States, proceeded Mr. Schorman, is the fluidity of oils. The University of Saskatchewan, he said, had placed its facilities at his disposal on a recent visit and he had been able to conduct a series of tests under sub-zero conditions. The engine would start quite readily on one oil, S.A.E. 10, at 12 deg. below zero at 315 consistometer shear; with a crank it would start at 470 shear, or —22 deg. A second oil would give normal starting at not less than —5 deg., while a third would make normal engine-starting possible only at 8 deg. above zero. One naphthene-base oil which met a pour test at —18 deg. did not, as a matter of fact, make starting possible below —15 deg. with a normal battery.

Present A.S.T.M. methods will have to be revised, the lecturer contended, especially with respect to the cold test. One man sent in a sample of oil from Oklahoma which was guaranteed to stand a pour test at 35 deg. below zero but at —12 deg. was so viscous that it took a service station 1 hr. to empty a 1-quart bottle.

In reply to a question, Mr. Schorman said that, while in Saskatoon, he drove for 10 miles or so one day with the temperature at 26 deg. below zero, stopping a few minutes here and here, driving for half an hour, and stopping again, keeping within the speed limit. The maximum temperature the crankcase registered was +3 deg. Under these circumstances, he commented, as much as 50 per cent of the power of the engine would be consumed by the shearing of the oil. On the following day, at 25 deg. below zero, a trip of 104 miles was taken, reaching a speed of 45 m.p.h., and under these conditions the temperature in the crankcase increased by just 100 deg., to 103. This showed that, for cold-weather driving, shear means a tremendous lot in what one is going to get out of a car.

Information Elicited by Questions

Questioned as to the distance which might be driven before oil is changed, Mr. Schorman remarked that it depends on whether the driver is willing to use dirty oil. It is claimed that some oils will last much longer than others, but the fact remains that all oils accumulate dust and dirt. If oil can be kept clean, it will last indefinitely. With respect to reclamation, the speaker said he would not like to trust this to the ordinary garage attendant, as partial distillation is necessary, which is a more or less dangerous procedure.

Asked if the addition of graphite to oil would improve it, Mr. Schorman said that graphite is in the nature of a filler; it fills the fine pores of a journal and produces a very smooth surface. The result is that the journal becomes so smooth that it cannot lubricate at all, as there are no crevices to carry the lubricant.

One member inquired whether there is any advantage in mixing castor-oil with mineral oil, to which Mr. Schorman replied that this is approaching a delicate subject. "I only want to say," he remarked, "that I have never in all my experience seen a vegetable or animal oil mix with mineral oil and remain in solution. That may be a broad statement, but I have never seen it done." Pure vegetable-oil, he added, is a splendid lubricant, because it has the flattest viscosity-curve; it has a very low viscosity-temperature.

A high-speed machine is easy to handle as regards oil, because the oil film is renewed in it more frequently at high velocity, explained Mr. Schorman. He would rather undertake to lubricate an engine running at 3000 to 4000 r.p.m. than one running at 800 to 900 r.p.m.

Experiments in Vaporizing Fuel

Buffalo Section Learns About Results Obtained with a Pre-Manifold Vaporizing-Device

SOME results obtained in internal-combustion engines in which practically all the vaporization was accomplished before the mixture entered the manifold system were recorded by Prof. George Autenreith and Prof. Harry Baum, of the College of the City of New York, in a paper entitled, Pre-Manifold Vaporization, which was read by the former at the technical session of the Buffalo Section held at the Statler Hotel, Buffalo, Dec. 3. The meeting, which was preceded by a dinner, was attended by 150 members and guests, and, under the chairmanship of William Edgar John, Secretary of the Section, the paper was followed by a prolonged discussion.

Following a brief résumé of the history of various means used to vaporize liquid fuel, Professor Autenreith stated that, if complete or approximately complete vaporization can be produced be-

(*Continued on page* 108)

Chronicle and Comment

Nominating Committees To Organize

THE ANNUAL Nominating Committee, composed of one representative chosen from and by each Section, and three members-at-large to be elected at the Business Session of the Annual Meeting, will organize during the Annual Meeting in Detroit this month. This Committee will send to the Secretary, on or before Sept. 1, the names of consenting nominees for the elective offices next falling vacant under the Constitution, except the Vice-Presidencies.

The nominees for the Vice-Presidencies will be selected by committees named by the respective Professional Activities. Under the new rules, the Vice-President of each Professional Activity each year is to be nominated by a Nominating Committee consisting of four Society members, elected at a stated Business Session of the Professional Activity which he is to represent, held prior to Aug. 1. Stated business sessions of six of the eight Professional Activities of the Society will be held during the Annual Meeting at the times mentioned in the program on p. 4.

Chicago World's Fair, 1933

IN 1933, the one hundredth anniversary of the settlement of Chicago will be celebrated by means of a World's Fair. Preliminary plans indicate that the exposition will illustrate a Century of Progress, with "the contributions of pure and applied science to industrial development during the last 100 years" as the main theme. An appeal has been made to the National Research Council for assistance in laying out a plan of presentation of such contributions, and in response to this appeal the Council appointed a group of representative men as a Science Advisory Committee.

The plan of operation provides that each member of the Advisory Committee shall become chairman of a subcommittee to be made up of men working in that particular field in which the chairman himself is engaged. The Society was asked to name the members of this Committee who should represent the automotive engineering interests, and C. F. Kettering was appointed. He has chosen, as the members of his subcommittee, the following:

Luther Bell, general manager, Aeronautical Chamber of Commerce, Inc.

F. O. Clements, technical director, General Motors Corp., Research Laboratories.

H. M. Crane, technical assistant to president, General Motors Corp.

W. J. Davidson, executive secretary, General Technical Committee, New Devices Committee, General Motors Corp.

H. H. Emmons, vice-president, Detroit Motorbus Co.

F. L. Faurote, consulting engineer.

A. H. G. Fokker, consulting engineer, Fokker Aircraft Corp.

C. B. Fritsche, president, Aircraft Development Corp.

G. A. Green, vice-president in charge of engineering, General Motors Truck Corp.

L. C. Hill, vice-president and general manager, Dietrich, Inc.

F. C. Horner, assistant to vice-president, General Motors Corp.

L. S. Horner, president, Niles-Bement-Pond Co.

K. T. Keller, vice-president and general manager, Chrysler Corp.

C. F. Kettering, general director, General Motors Corp. Research Laboratories.

O. T. Kreusser, director, General Motors Proving Ground.

S. A. Miles, National Automobile Chamber of Commerce.

Alfred Reeves, general manager, National Automobile Chamber of Commerce.

F. B. Rentschler, Pratt & Whitney Aircraft Co.

Jay W. Smith, Chris-Craft Marine Motor Co.

H. R. Sutphen, vice-president, Submarine Boat Corp.

W. R. Strickland, assistant chief engineer, Cadillac Motor Car Co.

O. D. Treiber, president and chief engineer, Treiber Diesel Engine Corp.

C. B. Veal, research manager, Society of Automotive Engineers, Inc.

Chance Vought, president, Chance Vought Corp.

W. G. Wall, consulting engineer.

Edward P. Warner, editor, *Aviation*, McGraw-Hill Publishing Co., Inc.

John Warner, assistant research engineer, Studebaker Corp.

F. A. Whitten, chief engineer, McCord Radiator & Mfg. Co.

J. F. Winchester, superintendent of motor-vehicles, Standard Oil Co. of N. J.

L. M. Woolson, aeronautical and research engineer, Packard Motor Car Co.

Mr. Veal has been made secretary of the subcommittee.

Mr. Kettering's subcommittee held its first meeting in Detroit on Dec. 5. At that meeting it was decided that the subcommittee should function in two divisions in working out a recommendation regarding the proposed exhibit. One division is composed of the men in the vicinity of Detroit; the other division, of the men in the vicinity of New York City.

Chairman Kettering plans to hold the next meeting of the subcommittee in New York City at the time of the Automobile Show, after which it will make a recommendation to the Science Advisory Committee regarding the type of exhibit that will best represent the automotive industry at the Chicago World's Fair.

Manly Awarded Langley Medal

THE Smithsonian Institution has awarded the Langley Medal for Aeronautics, posthumously, to the late Charles Matthews Manly, in recognition of his pioneer developments in aviation. The award was made by the Board of Regents at their annual meeting, held Dec. 12, 1929, Chancellor William Howard Taft presiding. The medal has been awarded to only six other men; in 1909 to Wilbur and Orville Wright; in 1913 to Glenn H. Curtiss and Gustave Eiffel; in 1927 to Col. Charles A. Lindbergh; and in 1929 to Commander Richard E. Byrd.

Mr. Manly was associated with Secretary S. P. Langley from 1898 to 1905. In 1902 he designed and built an engine developing 52.4 hp. for a weight of 125 lb., or a weight of 2.4 lb. per hp.—an epochal achievement.

Wings—a Coordinated System of Basic Design

By RALPH H. UPSON[1]

CLEVELAND AERONAUTIC MEETING PAPER — *Illustrated with* CHARTS AND DRAWING

TAKING as basic requirements such fundamental characteristics as can be largely separated from the problem of wing design or assumed as attributes of the complete airplane, the author discusses the independent variables consisting mainly of the geometrical characteristics that can be varied to obtain maximum performance without changing the basic requirements. He develops a weight and a drag equation, each founded on the chosen basic requirements and including in the simplest possible form the combined effect of the independent variables.

The terms in these equations are defined and the equations are applied to a low-wing monoplane in power and gliding fight and to a rectangular-wing biplane in gliding flight. The results are tabulated in some instances but are principally shown on charts. The accuracy of the results obtained is stated to depend largely upon the proper choice of approximations with an appreciation of their limitations. Certain reservations as to the accuracy of the equations are stated. The reduction to practice of the drag equation is discussed, and this is followed by comments on wind-tunnel research and flight-test data. Interesting points, bearing directly on practical design and having general application throughout the particular ground covered in the paper, are presented. Various contributing and supplementary data are given in appendixes.

THE purpose of this paper is to present a practical coordination of existing data as a foundation for judgment in design and research. Though the paper itself necessarily involves considerable recourse to engineering judgment, it is believed to furnish a much improved method for establishing the best general proportions of wings.

To avoid that loose thinking so often associated with artificial criteria of merit or performance, let us start with what we want the airplane for, taking as basic requirements such fundamental characteristics as can be largely separated from the problem of wing design or assumed as attributes of the airplane as a whole.

Basic Requirements

Fixed Load.—This may be defined in various ways, depending on how much is preconceived. The so-called pay-load may be included without argument. But accommodations have to be provided for holding it. Someone has to fly it. Means for starting and landing must be provided. The powerplant, though theoretically variable, is usually picked in advance. Even the capacity of the fuel tanks is usually regulated more by hours than by distance. For present purposes let us therefore assume the entire fuselage and contents, together with the landing-gear, as fixed, except such part of the tail and tail surfaces as is chargeable to the wing characteristics.

Stability and Control.—These are also taken as basic requirements because they are largely outside the scope of the present paper. The final choice of an airfoil must, of course, include consideration of the center-of-pressure travel, but for preliminary analysis the weight and drag of the stabilizing and control system will be included in the various terms representing the weight and drag of the wings, the estimated proportion being somewhat dependent on the general type of construction. Similarly, such items as dihedral and sweep-back are details to be determined later by conventional methods.

Structure.—The general type or efficiency of structure is assumed and the weight of its main elements adjusted to suit the geometrical characteristics of the wings, keeping the load factor at a figure sufficient to take care of stiffness as well as strength. To avoid too much variety of structure, most of the examples in the paper will be single-engine monoplanes, which form the best foundation for extension to other types. The high angle of attack will be the general criterion for strength, the other conditions being tentatively assumed to be proportional to it or included in it. We are not concerned here with detail stress-analyses but only with the importance of structural considerations in general wing design, particularly the effect on drag.

Fixed Drag.—This is essentially the direct drag of everything represented by the fixed load. Its separate identity is not so well marked, however, as that of the weight. It may be defined more specifically as that portion of the total drag which remains substantially constant for any one speed throughout the possible range of wing variation and varies directly as the square of the speed. Certain interference-effects will be treated separately because they have a direct bearing on wing design, particularly in their reaction to varying speed.

Speed.—The general drag formula will be worked out for the one standard speed of 100 m.p.h. The results for other speeds can then be tabulated or plotted over the expected speed range by a process of simple proportion. However, the basis for these speeds should be carefully noted. In the preliminary analysis covered by the basic formula, each speed stands alone as if the angle and median camber were laid out to suit that speed and no other. Broadly speaking, then, the drag curve of the actual airplane may be assumed to match the general curve at one speed only, though actually the agreement for a good airfoil-section is very close over

[1] M.S.A.E.—Aeronautical engineer, Red Bank, N. J.

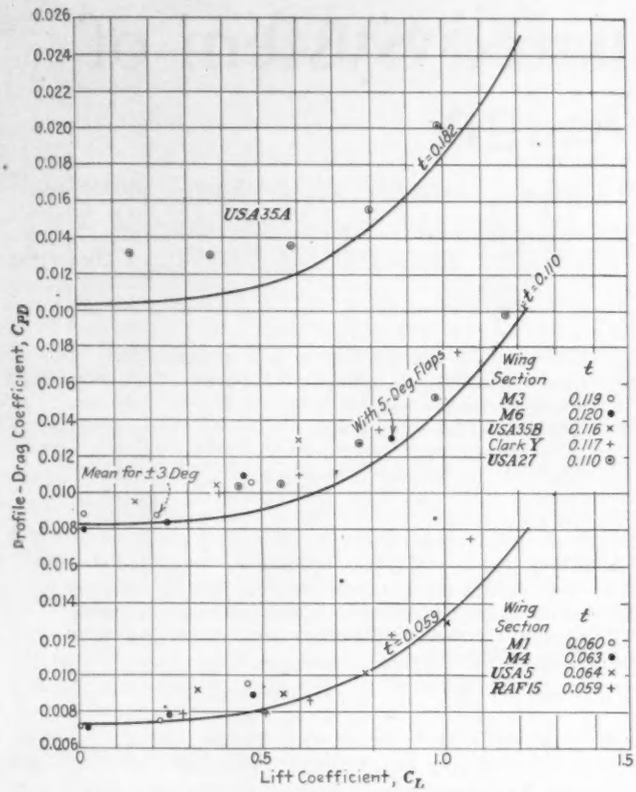


FIG. 1—PROFILE DRAG FOR A NUMBER OF WING SECTIONS AT A REYNOLDS NUMBER OF APPROXIMATELY 3.5×10^6

The Test Data Have Been Corrected for Tunnel-Wall Effect but Not for End Drag. The Profile-Drag Curves Are Plotted from the Formula

$$C_{PD}=(0.007+0.1t^2)\cdot(1+0.8C_L)$$

where

C_L = Lift Coefficient
C_{PD} = Profile-Drag Coefficient
t = Thickness/Chord

a wide range of speed. The maximum speed and stalling speed are treated not as basic requirements but as derived characteristics, the stalling speed being usually approximated in advance by consideration of the wing-loading.

Independent Variables

We come now to the independent variables consisting mainly of geometrical characteristics that can be varied and compromised one with another to obtain maximum performance, without changing what we have chosen to call the basic requirements.

Wing Area.—This is taken as the exposed plan-form area outside the fuselage according to usual American practice, the portion within the fuselage being added separately when necessary, as in the case of the effective lifting-area. The tail surfaces, although assumed to be proportional to the wing area, are not added directly but taken care of in the magnitude of the coefficients.

For structural purposes, the loading over the wing is assumed constant, neglecting fuselage lift and wing-tip losses except in special cases. Both these assumptions are on the safe side, the first being most severe on tapered wings and the second on rectangular wings.

Plan Form.—This is represented by four component variables, as follows:

(1) *Span* from tip to tip

(2) *Cantilever arm*, tentatively taken as the distance from side of fuselage, or root of cantilever, out to lateral center of area of one wing, or such area as constitutes the cantilever portion

(3) *Root chord*, the chord length adjacent to the fuselage

(4) *Plan-form factor for induced drag* (See Appendix 3) which, being of small effect in most cases, is included in the more general induced-drag factor as used in the complete formula

Front Elevation.—This will be determined by the section thickness except for the extreme tips, which are assumed to be faired sufficiently to eliminate the tip drag (See Appendix 2). Dihedral angle is ignored as far as any effect on weight and lift is concerned. Where outside struts are used their weight and drag are figured separately from the wing structure proper. The latter is assumed to have its critical points, in shear and bending, at the outer strut connections.

Wing Section.—This will be tentatively represented by *root thickness* alone; and except where otherwise stated, the thickness ratio (thickness/chord) is assumed constant over the entire wing. The median camber and other geometrical details of the wing section are derived dimensions, as already indicated. By this device of establishing the aerodynamic characteristics of the required wing-section before its geometrical proportions, except thickness, the otherwise somewhat complicated task of picking an actual section may be postponed until everything else is practically settled. The spar depth is assumed proportional to the wing thickness at maximum section minus the effective flange-depth.

The purpose of all the foregoing is to establish a reasonable and straightforward basis for a weight and a drag equation, each founded on the chosen basic requirements and including in the simplest possible form the combined effect of the various independent variables.

Dependent Variables

Weight.—The structural constants in the seven-term weight equation (1) are chosen purely as a result of practical experience, static tests and engineering judgment. When more efficient structures are developed, the result will be reflected in the numerical value of these constants. They will also change when the type of construction changes, as it will for marked departures in size. Very large airplanes may also call for a more refined equation with additional terms. But in any case the important thing is to distribute the *entire* gross weight in proper proportion among whatever terms are used. The resulting estimates may still differ considerably from the actual final weight, but the latter is of secondary importance if we have the best obtainable combination of weight, drag and other characteristics.

Drag.—This is now the main dependent variable, including the effect of the total weight. Like the weight equation, the drag equation (2) also has seven terms, but no direct term-for-term relation between them exists. The coefficients, although variable for different types of airplane, rest on a more absolute basis than in the case of the weight. In general, the equation is based on proved aerodynamic theory, with coefficients adjusted to suit actual conditions and the gaps filled in by rational empirical relations from the best data available. The lat-

TABLE 1—DRAG COEFFICIENTS

Term	Coefficient for 100 M.P.H.	Drag, Lb., for Speed of 100 M.P.H.	80 M.P.H.	60 M.P.H.	50 M.P.H.	Remarks
D_1	$qH = 125$	125	80			Balance from D. For power flight only. Add propeller drag for gliding.
D_2	$qJ = 0$	0		66	46	No outside bracing.
D_3	$qN = 0.22$	43	38			Smooth skin.
D_4	$qP = 3.3$	16				Faired tips.
D_5	$Q = 0$	0	0	0	0	Rounded nose and open cockpit.
D_6	$R/q = 0.014$	15	24	42	61	For power flight only. Probably 40 per cent greater for gliding.
D_7	$U/q^2 = 0.000012$	1	2	5	10	$D_3 + D_4 = 59$
D		200	144	113	117	Approximated from test at 80 m.p.h. and plotted as curve B in Fig. 4.
C_D		0.037	0.042	0.058	0.087	
C_L		0.247	0.385	0.685	0.988	

ter have been taken almost exclusively from the National Advisory Committee for Aeronautics variable-density-tunnel or full-scale flight-tests. (See Appendix 1 for details).

In the drag equation (2), it will be noted that terms D_3, D_4 and D_7 together form the *profile drag*. The addition of D_6 completes the terms necessary for an examination of the airfoil by itself. Expressed in terms of absolute or National Advisory Committee for Aeronautics coefficients for smooth airfoils of uniform thickness ratio, the over-all drag coefficient

$$C = C_D{}^2 (1 + \delta)/\pi n + (0.007 + 0.1t^2) \cdot (1 + 0.8 C_L{}^3)$$

based on tests at a Reynolds number of about 3,500,000. (See Fig. 1 and Appendixes 1, 2 and 3.)

The *parasite drag* and interference, except of tail surfaces, might be considered as the sum of the remaining three terms, or, expressed as a coefficient based on effective wing area,

$$C_{par} = (H/S_L) + (J/S_L) + (QW/qS_L{}^2)$$

The airplane drag is *not* the sum of the airfoil drag plus the separately measured parasite drag. Anyone who jumps to the conclusion that it is can easily be in error by over 20 per cent, as will be shown by numerical examples. Hence we must get back to the general equations. As might be expected, these form no easy short-cut for an otherwise incompetent designer. Rather, they are tools that, with proper knowledge and judgment, can be made to give better and more-certain results than any less-systematic procedure. This is equally true whether they be employed for the planning of new research and design, for the interpretation of precise wind-tunnel and flight-test data, or as a yardstick by which the ordinary performance facts available to any manufacturer can be compared and judged. Let us first take a practical example in this last class.

[2] See propeller-efficiency and power-available charts in Engineering Aerodynamics, by Walter S. Diehl.

Low-Wing Monoplane in Power Flight

Here I shall take the airplane with which I am at present most familiar, the Aeromarine Klemm AKL-25, Fig. 2. This has full-cantilever wings tapered in thickness ratio as well as in plan, with wood box-spars and plywood skin. Necessary dimensions are approximately as follows:

$a = 1.19$ ft., $(a/c)_m = 0.16$, $b = 40$ ft., $e = 6.55$ ft., $d = 0.19$ ft., $e = 2.1$ ft., $S = S_C = 196$ sq. ft., $S_L = S + ce = 210$ sq. ft., $\hat{y} = 8.88$ ft.

An estimate of the present *weights* in pounds and their corresponding coefficients are as follows:

W_1	W_2	W_3	W_4	W_5	W_6	W_7	W_8
	0[b]	157	84	40	44	0[c]	1,325[d]
1,000[a]		A	BW_1	EW_1	FW_1		
		0.80	0.24	1.2	1.0		

[a] Approximate.
[b] No outside bracing.
[c] Carried by plywood skin.
[d] Gross flight-load.

In the *drag* estimates given in Table 1 the coefficients for all the terms but D are first established for q_{100} at 100 m.p.h.; then for q_{80} at 80 m.p.h.; which equals $0.64 p_{100}$; and similarly for other speeds. D is figured approximately from the engine and propeller characteristics, as checked by flight tests, and D_1 as a balance on the other terms[2].

As will be noted from Table 1, the drag items fall

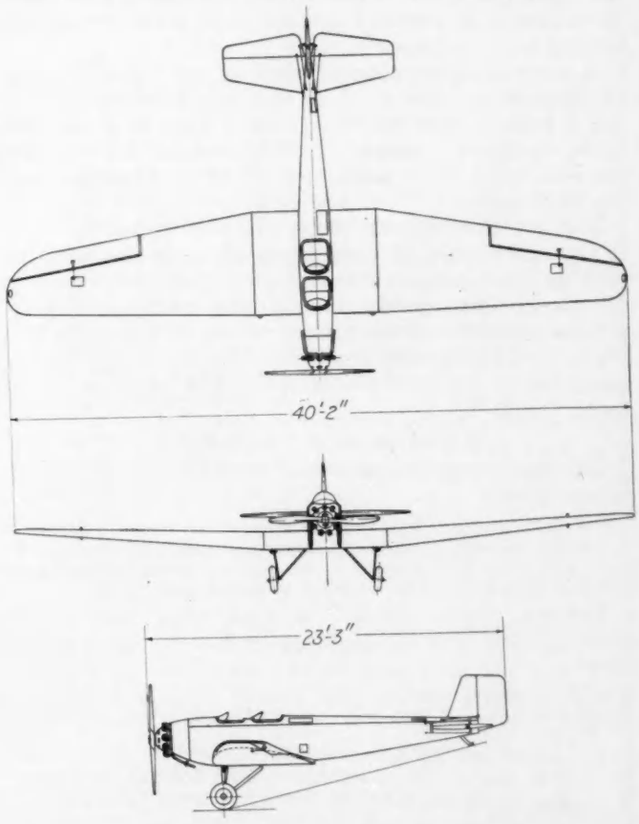


FIG. 2—PLAN VIEW AND FRONT AND SIDE ELEVATIONS OF THE AEROMARINE KLEMM AKL-25 AIRPLANE

Assumed Fixed Load=1000 Lb.; Fixed Drag=125 Lb. at 100 M.P.H. in Power Flight

Summary of Wing Variables

a = Maximum thickness at root of cantilever in feet
a/c = Root-thickness ratio
$(a/c)_m$ = Mean thickness ratio. For wings tapered in plan only, this ratio equals a/c
b = Over-all span in feet
c = Chord at fuselage in feet
d = Mean difference in feet between total span depth and effective spar depth between flange centers at root, increased by the ratio of maximum wing thickness to total spar depth
e = Width occupied by fuselage in feet
$n = b^2/S_L$ = Aspect ratio
q = Maximum dynamic pressure in pounds per square foot
S = Plan-form wing area in square feet
S_C = Cantilever area in square feet
S_F = Plan area of fuselage and struts in square feet
S_L = Plan-form wing area in square feet, corrected to include effective equivalent lifting-area contributed by fuselage and struts
S_C/S = Ratio of cantilever area to total wing area. For cantilever wings attached directly to the fuselage, the value of this ratio is 1
$(S + ce)/b$ = Mean chord
$[(S + ce)/b] \times (a/c)_m$ = Mean thickness along span
t = Thickness ratio at any point y. For wings tapered in plan only, this ratio equals a/c
x = Chord of any given wing section
y = Lateral distance in feet of any given wing section from the root
$\hat{y}$ = Distance in feet of lateral center of area from root. For uniform straight taper, in plan only,

$$\hat{y} = (b - e)\,\frac{1 - 3K^2 + 2K^3}{6(1 - K - K^2 + K^3)}$$

where K, which is the ratio of the tip chord to the root chord, except where used to represent Munk's span factor for biplanes, equals $(2S/bc) - 1$.
$1 + \delta$ = Plan-form factor for induced drag.

into four main groups. The first, comprising D_1, D_2, D_3 and D_4, varies as q or v^2. The second, D_5, is constant throughout the speed range. The third, D_6, varies inversely as q or v^2. The fourth, D_7, varies inversely as q^2 or v^4. Thus, when the various drag terms are known for some one speed, those corresponding to any other speed can be determined quickly, often with but a single setting on the slide-rule.

A more accurate determination of the exponents will probably show most of them to have fractional values; but I believe that the values used here will continue to be employed because of their greater convenience, the difference being taken care of by an adjustment of the coefficients.

For new design work, we are interested, not so much in the exact effect of varying speed, as in the effect of varying wing proportions and other design changes at certain assumed speeds. For further study in this direction, we shall assume speeds of 100, 80 and 60 m.p.h., which for the assumed airplane might be arbitrarily labeled base speed, cruising speed and climbing speed respectively.

Variation with Taper Ratio

The following are assumed to be constant at the values given:

b = 40 ft., D_1 = 125 lb. at 100 m.p.h., e = 2 ft., S = 190 sq. ft., $S_L = S + ce$ + 1/2 wing-strut area (assumed), W_1 = 1000 lb. and $[\hat{y}S_C/(a - 0.2)S] = 10$. This keeps the maximum flange load and torsional stiffness constant at a reasonable figure.

Assume wings tapered in plan only, with edges straight from root to tip. Then, if K = tip chord / root chord,

$$ce = [S/(b - e)] \cdot [(2 \times 2)/(1 + K)] = 20/(1 + K)$$

$$(S_C/S)\hat{y} = 10(a - 0.2) = 38[(1 - 3K^2 + 2K^3)/6(1 - K - K^2 + K^3)]$$

$$(a/c)_m = a/c = \frac{(\hat{y}\frac{S_C}{S} + 2)(1 + K)}{100}$$

$$n = 1600/(190 + ce)$$

Weight and drag coefficients the same as before except as noted.

Each drag term in Table 2 is given for the three speeds of 100, 80 and 60 m.p.h., respectively, and the results are plotted in Fig. 3.

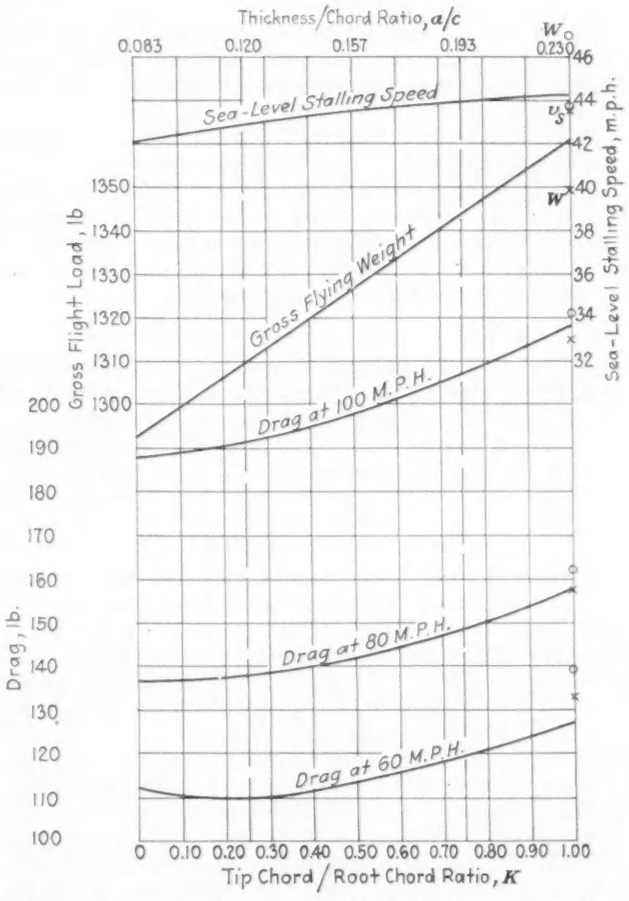


FIG. 3—CURVES OF SEA-LEVEL STALLING SPEED, GROSS FLYING LOAD AND DRAG FOR AN AIRPLANE SIMILAR TO THAT IN FIG. 2, HAVING AN OVER-ALL SPAN OF 40 FT. AND A WING AREA OF 190 SQ. FT., VARYING IN TAPER

Weight Equation

$$W = W_1 + W_2 + W_3 + W_4 + W_5 + W_6 + W_7 \quad (1)$$
$$= W_1 + W_2 + AS + B\,W_1\,[b\hat{y}\,Sc/(a-d)S] + E\,W_1\,(S+ce)\cdot(Sc/S)\cdot(a/c)_m + FW_1\,[(S+ce)^2/b^2](a/c)_m + GW_1(\hat{y}\,SSc)/(a-d)\,S$$

in which

A, B, E, F and G are constants depending on material and type of construction.

W is the *gross weight*.

W_1 is the *fixed load* as already explained.

W_2 is the weight of *outside wing struts*, wires and connections, if any. The best size will be determined to suit the individual case.

W_3 includes *skin or covering* of wings and tail surfaces and outside flanges or ribs and other stiffeners whose total weight depends primarily on wing area. For a structural skin, A may be reduced by whatever proportion acts as a useful part of the spar flanges.

W_4 includes *flange members* of all transverse beams or spars and any fittings carrying flange stresses. Basis: Conventional beam theory, assuming average weight per foot proportional to root weight per foot, for full cantilever wings. For externally braced wings the coefficient B is increased in the approximate ratio of mean flange stress over the entire span divided by the mean flange stress over the cantilever portions.

W_5 includes vertical transverse or spanwise *web or lattice members* of all beams or spars. Basis: Assume average web-weight per square foot proportional to root web-weight per square foot, the latter being assumed proportional to the shear load carried.

W_6 includes longitudinal *rib webs* and connection fittings. Basis: Assume uniform rib-spacing and uniform section-size and arrangement of lattice members, their total weight then being proportional to the load carried and to the wing section area at the position of mean chord. This holds good only for small airplanes.

W_7 includes the so-called drag bracing that really is *wing-torque bracing* in most monoplanes. Basis: With angular deflection at point $\hat{y}$ as the criterion, assume all torque load carried in the top and bottom trusses, the average weight of bracing per square foot of wing area being proportional to the root weight per square foot. For a structural skin, G is usually zero.

Drag Equation

$$D = D_1 + D_2 + D_3 + D_4 + D_5 + D_6 + D_7 \quad (2)$$
$$= qH + qJ + qNS + qPS(a/c)^2_m + Q\,(W/S_L) + (R/q)\cdot(W/b)^2 + (U/q^2)\cdot(W^3/S_L^2)$$

where

qH, qJ, qN, qP, Q, (R/q) and (U/q^2) are coefficients depending on speed, air density, scale and type and arrangement of aerodynamic elements. In the following description of terms the values of the coefficients are indicated for a Reynolds number of about 5×10^6, which is equivalent to a chord of 6.4 ft. at 100 m.p.h., a speed of 100 m.p.h., and standard sea-level density, 0.0765 lb. per cu. ft., making $q = 25.6$ lb. per sq. ft.

D is the *total drag* at the given speed.

D_1 or qH is the *fixed drag*, as already explained.

D_2 or qJ is the direct drag added by *outside struts* and connections, if any. The added drag due to angle of attack is included in D_5.

D_3 is essentially *skin friction* or that part of the profile drag that is substantially constant per unit surface. Basis: Skin friction theory. Corrected for scale, qN is 0.17 for a smooth airfoil alone; about 0.22 to 0.24 for practical wings including tail surfaces, or 0.27 if continuously corrugated. With loose fabric, qN may be much higher, but quantitative tests are needed here.

D_4 is the added profile drag due to *thickness ratio*. Here $S\,(a/c)_m^2$ is most properly given by the expression $2\int_0^{(b-e)/2} t^2\,x\,dy$ making $(a/c)_m$ slightly larger than the thickness ratio at point $\hat{y}$ for straight taper. Basis: Empirical. Here $(qP)_{100}$ is 2.6 for airfoil alone; about 3.3 to 3.5 including tail surfaces, or 4.0 for continuous corrugations.

D_5 is the approximate added drag, due to angle of attack of fuselage and other parasite parts, usually termed *variable parasite*. Basis: Empirical. The straight-line variation is simply assumed. For a low-wing monoplane, or any fuselage with a square-section nose, Q will be slightly positive, unless reduced again by the presence of open cockpits. Usual maximum is probably $0.03S_F$. In certain cases it may be negative, notably in a parasol type of airplane with interference between the fuselage and wing.

D_6 is the superficially well-known *induced drag* created by the load itself and the manner of its distribution by the wings. Basis: Theory of Prandtl. For a simple elliptical airfoil $(R/q)_{100} = 0.0124$. For the actual airplane with tail reaction and other unavoidable irregularities of load distribution, R/q can hardly be less than 0.014 and can apparently run up to 0.023 if an unrelieved spoiler effect at mid-span is present.

D_7 is the additional profile drag due to the magnitude of the *lift coefficient*. Basis: Empirical. For most cases the coefficient U/q^2 can be safely put as a constant, but indications are that a broader field is served by making $U/q^2 = 0.00004\,(D_3 + D_4)/S$, as it is then practically independent of changes in other design conditions.

Effect of Outside Struts

Assume now that the *rectangular wings* ($K = 1$) are braced on top by four 2-in. struts, two on each side extending half way out on the wing. Then $\hat{y}$ is cut in half, $S_L = 210$ sq. ft. and $S_C/S = 0.5$, $W_2 = 104$ lb., $D_2 = 23$ lb. at 100 m.p.h. and we have the following results for Case 1:

	Case 1			Case 2		
K	1.0			1.0		
ce	10			10		
a/c	0.088			0.088		
$W_1 + W_2 + W_3 + W_4$	1,352			1,316		
W_5	11			11		
W_6	22			22		
W	1,385			1,349		
$D_1 + D_2 + D_3$	190	125	70	185	122	68
D_4	5			5		
D_5	3	3	3	2	2	2
D_6	22[e]	34	61	21[e]	33	58
D_7	1	1	5	1	1	5
D	221	163	139	215	158	133
n		7.6			7.8	
n/t		88			89	

[e] Multiplied by an estimated strut-interference factor of 1.25. For struts below the wing this would not apply.

The second case, using 3.445 x 1.325-in. streamline tubing, with each strut supported at the center by a vertical secondary brace, which gives $S_L = 205$ sq. ft., $W_2 = 68$ lb. and $D_2 = 18$ lb., is plotted as Curve *A* in Fig. 4.

Variation with Span

Going back to the full-cantilever wings and taking the case of $K = 0.25$, hold $S = 190$ sq. ft. Then

$$y/(a - 0.2) = 0.2(b - 2)/(a - 0.2) = 10 \text{ as before}$$
$$ce = 608/(b - 2)$$
$$a/c = (b - 2) \cdot (b + 8)/15{,}200$$

The detail figures are here omitted, the results being plotted in Fig. 5.

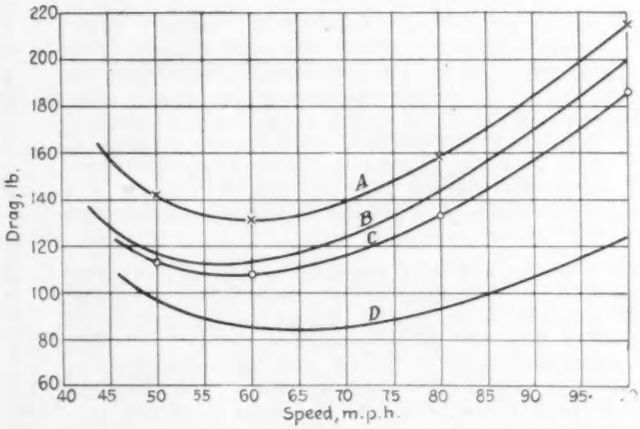


FIG. 4—SPEED-DRAG CURVES FOR FOUR SIMILAR AIRPLANES

Each Set of Wings Was Assumed To Carry a Fixed Load of 1000 Lb. The Data on the Different Wings Are as Follows: Curve *A*—Externally Braced Rectangular Wing Having a Gross Flying Load of 1349 Lb., Based on Points Plotted in Fig. 3; Curve *B*—Original AKL-25 Wing Having a Wing-Area of 196 Sq. Ft. and a Gross Flying Load of 1325 Lb.; Curve *C*—Wing Having a 4-to-1 Taper, a Plan-Form Area of 160 Sq. Ft. and a Gross Flying Load of 1281 Lb., Based on Points Plotted in Fig. 5; Curve *D*—The Same Wing as the Last with the Fixed Drag Reduced 50 Per Cent and an Assumed Gross Flying Load of 1281 Lb. The Lower End of Each Curve Is the Stalling Speed

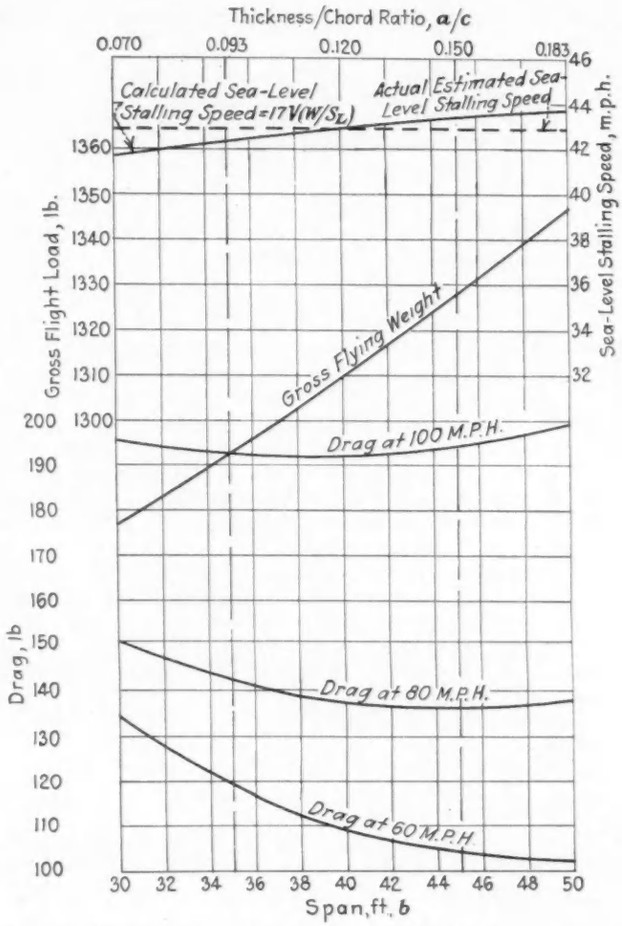


FIG. 5—CURVES FOR A WING AREA OF 190 SQ. FT. AND A TIP-CHORD/ROOT-CHORD RATIO OF 0.25, SHOWING THE VARIATION IN SEA-LEVEL STALLING SPEED, GROSS FLYING LOAD AND DRAG WITH DIFFERENT SPANS

The Assumed Fixed Load = 1000 Lb.; Fixed Drag = 125 Lb. at 100 M.P.H.

Variation with Area and Chord

Returning to a *constant 40-ft. span*, vary the area by the chord alone, keeping the thickness constant at 0.96 ft., and $K = 0.25$. Then $ce = 0.0843S$ and $a/c = 22.8/S$. The results are plotted in Fig. 6, and the case of $S = 160$ also transcribed to Fig. 4 as Curve *C*.

Variation with Thickness

All other dimensions being constant, $S = 190$ sq. ft., $b = 40$ ft. and $ce = 16$; vary a/c (See Fig. 7). Although W_7 is assumed to be zero throughout, a thickness ratio, a/c, below 0.12 probably will require special torque bracing or thicker skin.

Effect of Reduction in Fixed Drag

Consider D_1 arbitrarily cut in half, other characteristics remaining as in Curve *C*, Fig. 4, where $S = 160$ sq. ft. The new curve is shown as Curve *D* in the same illustration. As to how the above reduction might be effected is, of course, largely a matter of detail design of fuselage, powerplant and landing-gear; but it also depends on proper coordination with the wings, so much so that all the effort expended to clean up a fuselage may be nullified by wing interference. As brought

TABLE 2—WEIGHT AND DRAG COEFFICIENTS AT VARIOUS SPEEDS

$\hat{y}$	K	ce	a/c	$W_1+W_2+W_4$	W_5	W_6	W	D_1+D_2	D_4	D_6	D_7	D	n	n/t
9.50	1.00	10	0.230	1,248	55	58	1,361	167	33	17[f]	1[g]	218		
								128		27	3	158	8.0	35
								72		47	8	127		
9.05	0.75	11	0.193	1,248	47	49	1,344	167	23	16	1	207		
								122		25	2	149	8.0	41
								63		45	7	120		
8.45	0.50	13	0.157	1,248	38	40	1,326	167	15	15	1	198		
								116		24	2	142	7.9	50
								66		42	6	114		
7.60	0.25	16	0.120	1,248	30	32	1,310	167	9	15	1	192		
								113		24	1	138	7.8	65
								63		41	5	109		
6.34	0.00	20	0.083	1,248	21	23	1,292	167	4	17[f]	0	188		
								110		26	1	137	7.6	92
								61		47	4	112		

[f] Multiplied by form factor 1.05 for $K=1.0$ and 1.15 for $K=0$.
[g] The coefficients for this column varied in approximate proportion to $(D_3+D_4)_{100}/S$.

out in the further examples and references, the following three conditions are confirmed trouble-makers:

(1) Any spoiler effect above or immediately in front of a wing. This is specially to be guarded against in a low-wing monoplane, high-wing cabin job or in the use of bracing above a top wing.

(2) A small gap between fuselage and wing, as in the conventional biplane or parasol monoplane. If above a top wing, it may also cause a spoiler effect.

(3) Acute reentrant angles, particularly above a wing and behind its maximum section; affecting especially the otherwise efficient round-section fuselage.[3] This is really another form of item (2).

Low-Wing Monoplane in Gliding Flight

The first of the foregoing three items will now be illustrated by reference to a most interesting test.

The British Aeronautical Research Committee published the results, in terms of net-area coefficients, of a very complete full-scale drag test on an early Junkers monoplane[4] having approximate dimensions as follows:

$(a/c)_m = 0.16$, $b = 48.5$ ft., $S = 378$ sq. ft., $S_F \approx 100$ sq. ft. and $S_L = S + ce/2^h = 400$

[h] Based on spoiler effect to be discussed later.

Using the values already given for maximum variable parasite and corrugated surface, fitting R/q to the observed curve and getting D_1 as a balance, we have the drag equation:

$$D_{100} = 280 + (0.27 \cdot 378) + [0.40 \cdot 378 \cdot (0.16)^2] + [3.0 \cdot (W/400)] + 0.023\,(W/48.5)^2 + [0.00004(0.27 + [0.4 \cdot (0.16)^2]) \cdot (W^3/400^2)]$$

Dividing throughout by qS, we get

$$C_D = 0.029 + 0.015 + 0.008C_L + 0.0950C^2_L + 0.0009C^3_L \quad (3)$$

The results are plotted cumulatively from left to right in Fig. 8.

The most interesting thing about Equation (3) is the extraordinarily high induced-drag, nearly double that given by the conventional method of taking $C_{ID} = (C^2_L\ S)/(\pi b^2) = 0.051\ C^2_L$ instead of $0.095\ C^2_L$ as above. The result of such a substitution is also shown in Fig. 8, by the dash line.

Such a discrepancy is evidently a result of the very severe spoiler effect produced by the fuselage. Knowing that the effect is there, it is readily visualized by considering the high nose radiator, the almost vertical

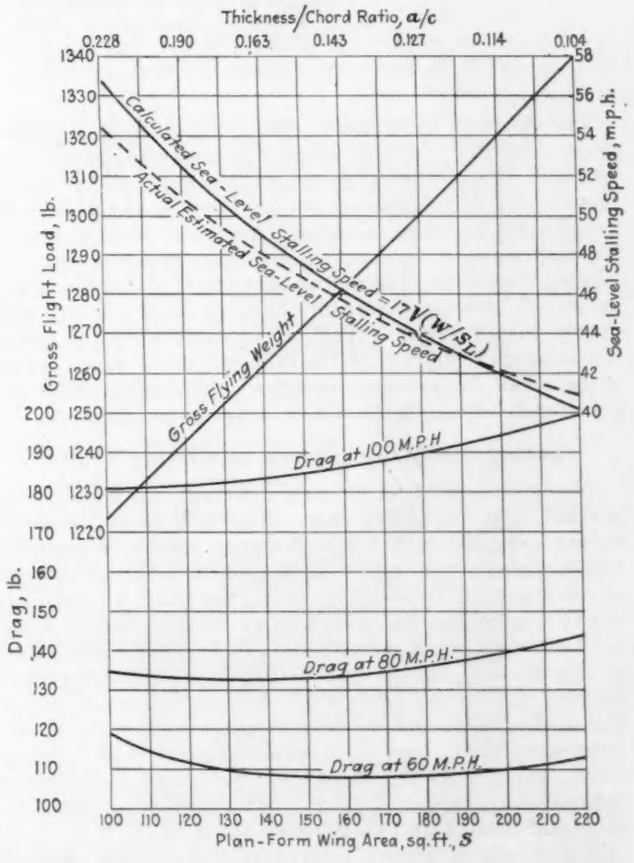


FIG. 6—CURVES SHOWING VARIATION IN AIRPLANE PROPERTIES WITH CHANGES IN PLAN-FORM WING-AREA AND CHORD

These Curves Are Based on a Span of 40 Ft. and Tip Chord/Root Chord=0.25; the Assumed Fixed Load=1000 Lb.; Fixed Drag=125 Lb. at 100 M.P.H.

[3] See National Advisory Committee for Aeronautics Technical Note No. 517.

[4] See Advisory Committee for Aeronautics (British) Reports and Memoranda No. 945.

TABLE 3—DRAG COEFFICIENTS FOR A RECTANGULAR-WING BIPLANE IN GLIDING FLIGHT

Term	Coefficient	Drag, Lb., for Speeds of 100 M.P.H.	80 M.P.H.	60 M.P.H.	50 M.P.H.	Remarks
D_1	qH	174	174			Balance from D
D_2	qJ					
D_3	$qN = 0.23$	34	134	76	53	Fabric covered
D_4	$qP = 3.5$	2				Squared tips
D_5	$Q = 0$					
D_6	$R/q = 0.0105$	27	42	75	107	From slope of curve
D_7	$U/q^2 = 0.000007$	0	1	3	6	
D		237	177	154	166	Gliding tests
C_D		0.062	0.073	0.113	0.175	
C_L		0.267	0.417	0.741	1.068	

windshield, and the large, open pilot's cockpit. But the two other points are at first hard to reconcile. These are

(1) Although these airplanes were notoriously weak on climb, they did not seem to be as bad as the measured drag would indicate.

(2) Although the AKL fuselage is obviously better, one would still expect to find considerable of this induced interference, a term suggested by J. D. North.[5] Laboratory tests show a noticeable effect even with a thin longitudinal plate above an airfoil, yet the data as given for the AKL-25 show no special evidence of extra induced-drag.

The effect on the Junkers was not the direct propeller drag, because that was measured separately and subtracted. The answer seems to be that, when under power, the slipstream tends to keep the air moving around the fuselage, so that it does not have a chance to stagnate and separate, which also explains why the gliding characteristics in the other case are not as good as would be expected from the climb and general efficiency under power. Another interesting point about the Junkers tests is that the induced interference seems to be about 30 per cent greater at full scale than for the model, although the direct drag as usual is less.

Rectangular-Wing Biplane in Gliding Flight

We come now to an entirely different but equally important type of interference. This will be represented by one example only, the well-known Sperry Messenger which, though now an obsolete type, has the advantage of having been thoroughly tested by the National Advisory Committee for Aeronautics. Four different wing sections were flight-tested, of which details are given in Report No. 220 of the Committee. Here we shall assume a fixed a/c of 0.064, which corresponds to the U.S.A.-5 airfoil, and a gross weight of 1010 lb. Other necessary dimensions are: actual span $b = 20$ ft., $c = 4.0$ ft., gap = 3.83 ft., $S \approx S_L = 148$ sq. ft. and biplane induced-drag factor, $1/K^2 = 0.75$ according to the conventional theory of Prandtl and Munk.

If virtually no spoiler interference is present $(R/q)_{100} = 0.75 \times 0.014 = 0.0105$. U/q^2 probably will be smaller in about the same proportion, that is, $(U/q^2)_{100} = 0.75$ $[0.0004(D_3 + D_4)/S]$. qN and qP are slightly increased to allow for the projecting ribs and the squared wing-tips. The other drag coefficients are obtained from the National Advisory Committee for Aeronautics glide tests, simply by choosing values that come nearest to fitting the experimental curves. The figures are presented in Table 3.

The principal interest in Table 3 lies in the combination $D_1 + D_2$ to see how near it comes to the fuselage drag measured separately. The National Advisory Committee for Aeronautics made a full-scale drag test on this same fuselage in the large tunnel, giving a net drag of 57.3 lb. at 80 m.p.h. for the complete fuselage and landing-gear, exclusive of tail surfaces. This would seem to make $(D_1)_{100} = 90$ lb. plus the drag of control horns and other distributed parts. But $D_1 + D_2$ from the glide tests, as given in Table 3, = 174 lb. No independent check was made on D_3, the wing bracing, but a rather liberal estimate makes it not more than about 30 lb. As this still leaves 144 lb. for D_1, we are forced to the conclusion that the interference and miscellaneous distributed drag was equal to about 60 per cent of the total measured drag of the fuselage and landing-gear. As the landing-gear, representing 45 per cent of the combination, is probably not much concerned in the interference effect, the above proposition can be restated as follows:

The drag of this fuselage plus interference and undetermined extras is more than double the drag of

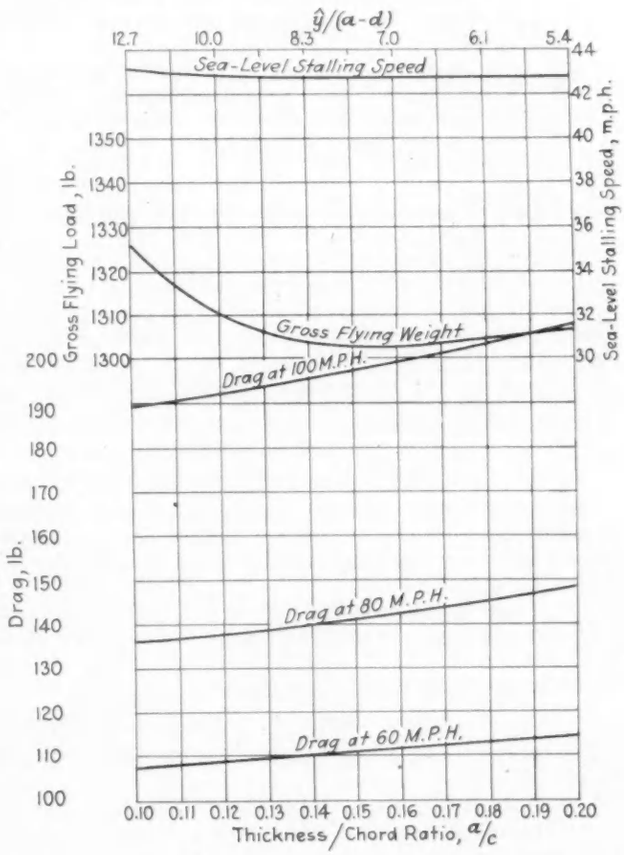


FIG. 7—CURVES SHOWING VARIATION IN AIRPLANE PROPERTIES WITH CHANGES IN THE THICKNESS

Based on Plan-Form Wing-Area=190 Sq. Ft.; Span=40 Ft. and Tip-Cord/Root-Chord Ratio=0.25

[5] See *Journal of the Royal Aeronautical Society*, January, 1929, p. 1.

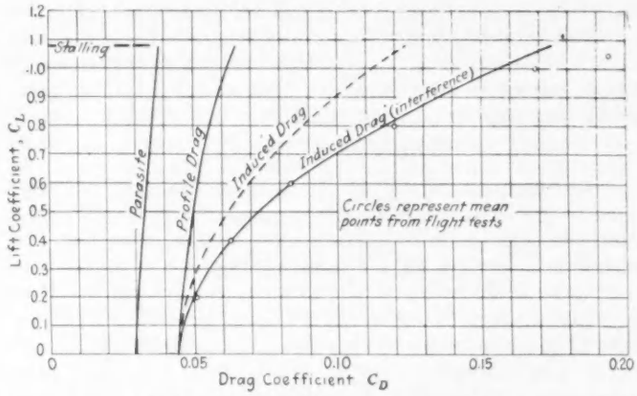


FIG. 8—EXAMPLE OF CUMULATIVE DRAG COMPONENTS IN TERMS OF THE LIFT AND DRAG COEFFICIENTS, SHOWING INDUCED INTERFERENCE

These Curves Are for a Large Low-Wing Monoplane in Gliding Flight

the same fuselage measured separately, including engine, windshield and other parts. (See Fig. 9).

Without appropriate tests, any further separation of the drag would be the merest guess. That the *direct interference* between the fuselage and the wings is the most important single item of drag in the entire airplane at the higher speeds seems at least probable. A model of this airplane was tested in the variable-density tunnel and, at a pressure equivalent to full scale, gave considerably lower drag-values than the flight tests. Aside from possible errors in the tunnel tests, this tunnel being new at the time, the most plausible explanation is the failure to reproduce the model exactly to scale, particularly in respect to the smaller parasite parts, most of which were on the fuselage, although an attempt was made to do so.

Performance

The foregoing examples should be enough to show the importance of dealing with such fundamentals as weight and drag in a way to make them easily negotiable in the preliminary stages of a new project. For any desired case, most of the performance characteristics are then simple functions of the weight, drag and the various assumed proportions. Although this is no textbook on airplane design, the following summary of the more fundamental relations may be found useful for reference:

(1) *Thrust horsepower required at sea level* $\approx (Dv_c/375) + v_z\ (W - T_z)/33{,}000$

where

D = the horizontal-drag value from horizontal flight for $v \approx v_c$

T_z = the vertical-thrust component

v_c = the resultant speed in miles per hour

v_z = the vertical rate of ascent in feet per minute

(2) For the *thrust horsepower available*, a good approximation can be had from the generalized curves given in Chapter 6 of Diehl's Engineering Aerodynamics. The same book gives methods of calculating the take-off and landing run from the weight, drag and other factors.

(3) *Gliding angle at sea level* $\Theta = \tan^{-1} D_g/W$ where D_g is the drag value from horizontal flight for a speed $v = v_g \vee \sec \Theta$ where v_g is the resultant gliding-speed under consideration. D_g is sometimes different from D, as already explained.

(4) *Stalling speed at sea level* $v_s = (19.77/C_{L\ max}) \vee (W/S_L) \approx 17 \vee (W/S_L)$ based on $C_{L\ max} = 1.35$ for preliminary calculations. For biplanes the coefficient 17 reduces to approximately 16.5[1].

(5) Effect of *change in air density from standard sea level*: Multiply v_s and all speed-scale values by $\vee (\rho_0/\rho)$ where ρ_0/ρ is the ratio of standard density to actual density. This does *not* apply to v_c and v_z in item (1), but only to the v scale on the drag chart from which D or D_g is taken.

Several systems of performance calculation use the full-scale polar as a basis. This can be readily constructed from the drag equation as shown in Equation (3), although preferably on a basis of S_L instead of S (See Appendix 7). A proper interpretation of these curves must now include reference to limitations imposed by the method itself.

Accuracy of Results

Speaking of results, not component parts, accuracy depends largely on the proper choice of usable approximations, with appreciation of how far they apply. The equations already given are subject to various reservations, including the following:

Errors of Commission

(1) The avoidance of fractional exponents, previously commented on.

(2) Scale effect for large departures from the assumed Reynolds number. The correction in the skin-friction coefficient is approximately ± 10 per cent for halving or doubling the Reynolds number respectively. The effect on the other coefficients is uncertain but probably small for the above range.

(3) Inaccuracies in the profile-drag coefficients due to insufficient experimental data (See Appendixes 1 to 4). The National Advisory Committee for Aeronautics has proposed a series of

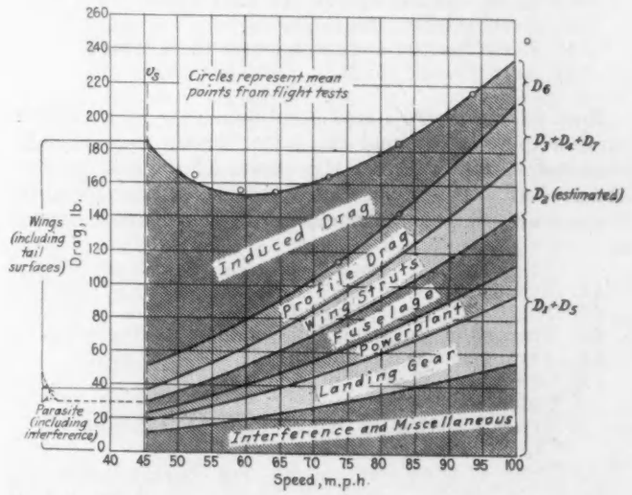


FIG. 9—EXAMPLE OF ITEMIZED VARIATION OF THE DRAG WITH SPEED, SHOWING DIRECT INTERFERENCE

These Curves Are for a Small Biplane in Gliding Flight. The Wing-Strut Drag Is Essentially Part of the Wing Drag

[1] See National Advisory Committee for Aeronautics Report No. 249 and Technical Note No. 310.

coordinated tests in the variable-density tunnel which should relieve this condition.

(4) Uncertain magnitude of the induced-drag coefficient for different conditions and arrangements of parasite parts. Further researches in a good atmospheric tunnel would throw great light on this important factor.

(5) The blanket conditions assumed to govern the variable parasite-drag; previously commented on.

(6) Assumptions as to the effective lifting-area; usually unimportant, but special cases may need special treatment (See Appendix 7).

(7) Approximation of the tail drag as an addition to the wing drag. This proportion must be changed to suit radically different conditions. The part changeable to the fuselage may be either plus or minus.

(8) Entire breakdown of drag formula at burble point. For cases where $C_{L max} = 1.2$ to 1.4, accuracy can hardly be expected beyond $C_L \approx 1.0$ where $v = 19.77\sqrt{(W/S_L)}$ for sea-level conditions.

(9) Generally approximate nature of the weight equation; good only for variations in degree from a practical example, as already mentioned.

Errors of Omission

(1) Ground effect on lift and drag; greatest on low-wing type

(2) Difference in induced drag with power on and off

(3) Other effects of propeller interference, usually included in efficiency

(4) Movement of center of pressure relative to center of gravity, depends on airfoil section, wing proportions and vertical position of center of gravity

(5) Maximum lift and true landing-speed, from airfoil section and wing proportions

(6) True profile-drag for a specific wing-section (See Appendix 1)

(7) Indirect structural increments due to aerodynamic changes, such as a reduction in load factors justified by improved stability or decreased maneuverability

(8) Greater wing stiffness required by an increase in speed and in ratio of aileron arm to span

(9) Beam stresses in ribs for thin wings

(10) Drag stresses for biplanes

(11) Extra leading-edge weight for thick wings

(12) Aerodynamic loads in fuselage

Most of the above items are taken care of in the reduction to practice, and the latter group in particular need not seriously affect the choice of general proportions. But entirely outside the scope of the formulas themselves, other limitations must be considered, such as:

(1) Production economy
(2) Practical span limitations
(3) Proportions favoring detachable or folding wings
(4) Arrangements for landing or flotation gear
(5) Utilization of internal wing-space
(6) Vision
(7) Accommodation of wing spars to fuselage requirements
(8) Effect of wing form on spinning characteristics

Any attempt at detail study of this last group of items would be beyond the purpose of this paper. For example, as between tapered and rectangular wings, the production advantage is certainly on the side of the latter. Whether it is worth the accompanying aerodynamic sacrifice is a question; but, as long as we know what this sacrifice is, we can judge accordingly.

[6] See Fluid Dynamics for Aircraft Designers, by Max M. Munk.

Reduction to Practice

As already explained, the drag equation does not pretend to give the complete drag curve of any specific airplane with a definite airfoil section but is rather the envelope curve of all sections having the chosen planform and thickness. The practical importance of this procedure must be apparent by now, in spite of errors that may be introduced thereby. That these errors are surprisingly small, however, can be seen from the profile-drag curves in Fig. 1, which shows that a good section tends to hug the envelope curve rather closely over a considerable range of lift coefficient. When the selection of general wing proportions has been narrowed down to a few individual cases, as in Fig. 4, substituting actual profile-drag for envelope profile-drag is simple. If the difference between the actual airfoil drag-coefficient and the empirical curve, Fig. 1, is ΔC_D, the difference in the drag from the general formula will be $\Delta D = S_q \Delta C_D$ at a speed of $v = 19.77\sqrt{(W/C_L S_L)}$ m.p.h. In Fig. 9, the difference is shown from the actual full-scale tests of the National Advisory Committee of Aeronautics, using the U.S.A.-5 wings.

The general principles guiding the choice of an airfoil section are simple enough. Any airfoil has two burble points or regions, an upper and a lower, (See Appendix 4). The ordinary flight-range must lie wholly between the two. A moderate increase in median camber tends to raise both of them. In general, choosing the largest possible camber that will not *appreciably* affect the cruising speed or *seriously* affect the top speed seems the best tentative approach. By so doing we get the largest maximum-lift that is feasible with a fixed airfoil, and, in addition, if the full-out lift-coefficient is not too low, the diving speed is checked by the lower burble-point that is reached just beyond the regular top speed. That the spinning characteristics will be of the best with such an arrangement is also probable, though not certain. The larger travel of the center of pressure, however, makes for larger tail-surfaces and heavier wing torque-bracing that must be taken into account before a final choice is made[6]. The slope of the lift curve is almost exclusively a function of the plan form, a convenient and closely approximate formula being given in Appendix 3.

In the future, I believe, airfoil sections, as well as general wing proportions, will be chosen mainly by formula. Some data on this are given in Appendix 5, using simple mathematical equations that, by a mere choice of constants, can be adapted to any desired thickness, curvature and position of the maximum ordinates. Unfortunately, the experimental data are still insufficient for checking the exact proper shape for a given thickness and range of lift, insofar as conditions near the burble point are concerned.

With results available from hundreds of different airfoils tested throughout the world, we ought to know something about airfoils, but most of these tests are practically worthless for purposes here considered. With a few notable exceptions, they are lacking in useful coordination, due apparently to insufficient theoretical study, *before* the tests, leading to some definite purpose.

Then the ordinary atmospheric-tunnel is so far out of scale as to make profile-drag and maximum-lift measurements very misleading.

From its position in both the weight and drag equations, thickness is evidently one of the fundamental factors. Although the curve of its net effect on the desired performance may be very flat over a certain range, its limitations should be determined so that judgment based on other considerations than weight and drag can be confined within the proper scope. In this the ordinary wind tunnel is wholly inadequate.

Wind-Tunnel Research

The foregoing remarks on the inadequacy of the small atmospheric-tunnel for airfoil investigations do not mean that such tunnels are of no use, but merely that the emphasis is changed. Instead of working on simple airfoils or on the direct determination of over-all characteristics, such tunnels should now be largely devoted to the study of interference and turbulent flow, including effects on lift, drag, stability and control.

For example, in the Sperry Messenger analysis given earlier in the paper I assumed that the apparent induced-drag was the true induced-drag. The latter, however, might really be somewhat larger, the difference being masked by a negative variable-parasite effect that sometimes accompanies a large direct-interference value. The two effects would be distinguishable only by reference to a change in aspect ratio. This could be done most accurately in full scale, but any good wind-tunnel probably would give a good qualitative indication. Plenty of other interesting tests will suggest themselves; for the road of research on interference is still practically open, the same as it was on airfoils in the early days.

To be sure, the bogy of scale effect is still on hand, but with one great difference: The airfoil problem is principally one of *kind*. Airfoils or wings we must have in some form; but *interference in each and every form should be, and presumably can be, virtually eliminated*, except when it can be made to serve a useful purpose. Where a quantitative check is needed, it can usually be had from simple flight-tests.

Flight Data

Examples of drag analyses from flight tests have already been given, but that these do not necessarily require an elaborate routine to serve a useful purpose should be clearly understood. D_3, D_4 and D_7 are determined in advance from the constants already given or from whatever more-accurate values may some time supersede them. If the nature of the design or wind-tunnel tests justify an assumption that the induced interference or variable parasite is not appreciable, D_5 and D_6 are also predetermined, and $D_1 + D_2$ is then simply the difference from the total drag at some one speed.

If either D_5 or D_6 have to be found, two speeds are necessary; if both, three different speeds. If either D_1 or D_5 is found to be unduly large, the assumed straight-line variation of the latter must be tentatively questioned. In this case, however, further tests to check off the design as a poor one are unnecessary.

Glide tests are probably the most accurate except for the absence of the slipstream effect, as noted earlier. For power flight all that is necessary is to check the number of engine revolutions and the airplane speed in level flight and at various rates of climb, *all at full throttle*, using the relations given in the section on Performance. The maximum rate of climb even without the air-speed, together with the maximum speed, will often tell the story. If desired, S_L can be checked by determining the actual stalling-speed of the airplane; then $S_L = 391\ (W/v_s^2 C_{L\ max})$ where $C_{L\ max}$ is from the airfoil tests *at full Reynolds number*. The importance of knowing the air density and of getting the true air-speed is, of course, obvious, but error in these respects would have to be very large to equal what is commonly introduced by unsound methods of performance analysis. (See Appendix 7.)

Some Scattered Conclusions

As indicated at the beginning, the object of this paper has not been so much to prove any specific points but to furnish the tools by which such can be attained. Nevertheless, for those who like to have their conclusions ready made, several interesting points bearing directly on practical design are incidentally brought out by this study. The following seem to have general application throughout the particular ground covered herein:

(1) The best taper, including both structural and aerodynamic considerations, makes the root chord about five times the tip chord. If a wing is to be tapered at all, little practical reason seemingly exists for a taper much less than this.

(2) Elliptical plan-forms, taper in thickness ratio or camber, and other fancy wing-forms possess no great advantages that cannot be closely approximated by a plain straight taper of correct proportions. Even tip fairings, admittedly of some importance for thick rectangular wings, cease to be of much consequence for a 5-to-1 taper.

(3) A set of internally braced wings of ideal proportions has from 20 to 30 per cent less drag, exclusive of that due to the fuselage, and slightly less weight than the best arrangement of rectangular wings, either internally or externally braced, except in a case where bracing with wire alone is possible. Considering also the improved stability and control afforded by tapered wings, the importance of devising economical production methods for such airplanes is apparent.

(4) The main diagonal struts of a conventional externally braced monoplane should each be supported by at least one intermediate stiffener. Such an arrangement is superior to a rectangular cantilever wing.

(5) Adequate span is the prime essential for any design with a large power-loading. Many small airplanes have been failures mainly because of inadequate span.

(6) If the span is maintained, the area can be varied through a considerable range without seriously affecting anything but the stalling speed.

(7) A cantilever wing with structural skin, for a light airplane, can often be made surprisingly thin, utilizing sections commonly thought to be applicable only to externally braced wings, provided any particularly harmful rate of vibration is not set up.

(8) The low-wing monoplane is attractive structurally and provides good general vision but tends to generate induced interference. By

correct design this can apparently be minimized and practically eliminated for power flight.

(9) Even an ideally streamlined fuselage can cause trouble if poorly placed in relation to the wings. An uncontrolled contraction of flow effect is especially to be avoided.

(10) Unless and until proved relatively harmless in specific cases, *any* gap between a fuselage or nacelle and a wing should be regarded with suspicion. This becomes of greater importance with the use of N. A. C. A. cowling. Many designs apparently need serious reviewing in consideration of the latter feature.

(11) The use of high-lift wing sections of fixed form depends on having an operating range at relatively high lift-coefficients, which is impossible in most cases in which landing speed is a controlling factor. It also depends upon having a relatively high aspect-ratio and vice versa. Thus an indirect, but not a direct, relation exists between wing thickness and maximum lift.

(12) For the wings and tail alone, if properly proportioned structurally, an L/D of 25 to 30 is about the maximum except for very special cases. (See Appendix 6.)

(13) Experience of the past year has shown beyond reasonable doubt that a *spin-proof airplane is entirely feasible*, merely by the right selection of proportions and balance. Although outside the general scope of this paper, it is worth noting that the required wing characteristics are mainly in the direction of improved general efficiency along lines herein suggested.

Summary

Isolated facts and opinions exist without end; in general, we already have an undigested surplus of both. The further possibility of radically new discoveries inherently precludes any attempt to discount them in advance. However, a real and directly available profit can be found in the refinement and better utilization of existing data. The sure profits lie here; all the rest is speculation.

Perhaps I have succeeded in scratching a little more deeply the surface of this important subject as applied to wings. If so, may it stimulate others to do some real digging, always remembering that the main object of performance analysis, whether in advance or in retrospect, is to *improve* the results, not merely to predict or determine them. For this purpose, the first requirement is a simple flexible system of coordinating the essential variables, such as is herein suggested and applied.

APPENDIX 1

Example of Profile-Drag Calculation U. S. A.-5 Airfoil

National Advisory Committee for Aeronautics—Report No. 233, Variable-Density Tunnel; Average Reynolds Number 3,630,000

Airfoil $b = 2.50$ ft., $c = 0.416$ ft., $n = 6.0$ and $t = 0.064$.

Closed-type tunnel: Diameter of throat, $d = 5.0$ ft. before reconstruction.

Tunnel-wall factor for induced drag $= 1 - \frac{1}{2}(b/d)^2 = 0.875$[j].

Plan-form factor for induced drag $= 1 + 0.009n = 1.054$. (See Appendix 3.)

Then

$$C_{ID} = 0.875 \times 1.054\ (C_L^2/\pi n) = 0.0489\ C_L^2$$

The total profile-drag, including the tip drag, is then $C_{PD} = C_D - C_{ID}$ in accordance with Table 4.

TABLE 4—PROFILE DRAG AT VARIOUS ANGLES OF ATTACK

Angle of Attack, Deg.	Coefficients: Lift, C_L	Drag, C_D	Induced Drag, C_{ID}	Profile Drag, C_{PD}
−3.0	0.101	0.0142	0.0005	0.0137
0.0	0.324	0.0144	0.0051	0.0093
+3.0	0.562	0.0244	0.0154	0.0090
6.0	0.782	0.0400	0.0299	0.0101
9.0	1.001	0.0629	0.0499	0.0130
12.0	1.188	0.0918	0.0690	0.0228

The importance of accuracy in the tests and calculations is apparent from Table 4. For example, ignoring the tunnel-wall correction for the 9-deg. angle of attack would make $C_{PD} = 0.0073$ instead of 0.0130, a difference of 44 per cent. For the same reason, supplementing the results thus far attained by additional tests in the variable-density tunnel, particularly on the thicker sections, is important.

APPENDIX 2

Tip Drag

The profile drag as generally used includes skin friction and turbulence effects that are mainly the function of the airfoil section and lift coefficient. It may also include an appreciable tip effect that is not part of the induced drag but represents, rather, a modification of the profile drag due to special conditions of flow at the tips; for example, consider a symmetrical section at zero angle of attack such as is shown on the left of Fig. 10. Such a section, if square at the ends, obviously will have a certain drag associated with the tips even with no lift whatever, and the effect is presumably of similar magnitude even when induced drag is present. This effect has been tested in the case of struts[7]. Though lacking a scale effect applicable to wings, it gives the general idea of the magnitude of the discrepancy from this source, which is probably negligible for thin sections but may run up to 10 per cent or more for the thicker sections. Accurate full-scale tests are needed on this point, particularly on varying aspect-ratios, so as to determine whether any other influence than the tip shape is of appreciable effect. The principal purpose would be not so much the direct application to wing design but rather to make possible a more-accurate interpretation of conventional tests on square-tipped airfoils. A rough estimate of the effect in question has been allowed for in the general profile-drag formula as given in Fig. 1, the formula being supposed to cover the sectional drag only and not the tip drag.

APPENDIX 3

Effect of Plan Form on Induced Drag and Lift Curve

The best treatment on the theory of this subject will be found in Aerofoil and Airscrew Theory, by H. Glauert. The theoretical factors, however, are slightly

[j] See Fluid Dynamics for Aircraft Designers, by Max M. Munk; also Engineering Aerodynamics, by Walter S. Diehl; also Aerofoil and Airscrew Theory, by H. Glauert.

[7] See National Advisory Committee for Aeronautics Report No. 137.

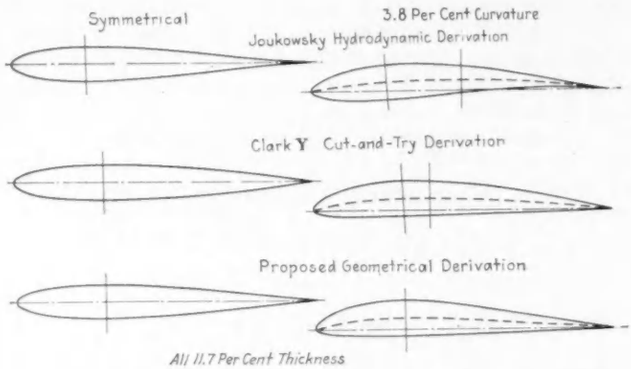


FIG. 10—EXAMPLES OF SYMMETRICAL AND CURVED AIRFOIL-SECTIONS HAVING THE SAME OVER-ALL DIMENSIONS BUT DIFFERING IN DERIVATION AND FULLNESS

modified by other influences in the case of actual airfoils. For the usual range of proportions, the following formulas give convenient approximations, the numerical values being chosen to suit best the results of the actual tests available to date.

Coefficient of induced drag, $C_{ID} = [C_L^2 (1+\delta)] / \pi n$

Increase of lift coefficient per degree, $dC_L/d\alpha = 0.1n / [n+1.9(1+r)]$ where $\delta = \delta_1 n$, $r = r_1 n$ and δ_1 and r_1 are given in Table 5.

TABLE 5—VALUE OF δ_1 AND r_1

K Tip Chord-Root Chord Ratio	δ_1	r_1
0.00	0.026	0.019
0.25	0.003	—0.005
0.50	0.002	—0.003
0.75	0.005	0.007
1.00	0.009	0.019

The above factors are accurate only for unwarped simple airfoils without interference or break in the flow. In the case of a fore-and-aft slot, the induced drag is so greatly increased that when such a slot has any appreciable width in proportion to the span, the effect approaches closely to the condition of two separate airfoils of one-half the aspect ratio[8]. As shown in the paper, a considerable portion of this effect may apparently accrue from a disturbance in the flow over the top alone.

APPENDIX 4

Maximum Lift and Lower Burble-Point

Usable data on these characteristics are still very meager, but an attempt is made in Fig. 11 to plot them on the basis of median camber alone, although other geometrical characteristics of the section should obviously be taken into account. In addition, the curve of lower burble-point is based on atmospheric-tunnel tests[9] from which at best no more than a qualitative idea can be had. Other things being equal, the lower burble-point should occur at a higher lift-coefficient and act as a greater check on the diving speed as the position of the maximum camber is moved forward, my attention being first called to this by Mr. Korvin-Kroukovsky. That the more-forward positions tend to minimize the center-of-pressure travel is also apparent.

[8] See Engineering Aerodynamics, by Walter S. Diehl.

[9] See National Advisory Committee for Aeronautics Technical Note No. 422.

Although the *shape of the median line* is thus a primary consideration in determining the burble points and the center-of-pressure or moment coefficient, it is not and cannot be properly so considered in respect to profile drag. As noted in the paper, the profile drag of a whole family of best sections depends only on thickness and lift coefficient. This gives a series of optimum profile-drag values from which a specific airfoil will epart only if its section shape, other than thickness, is unsuited to the particular combination of thickness and lift coefficient. Considering the profile in relation to the median line, the geometric character of the best profile probably will vary mainly with thickness, if at all, and that of the median line will vary mainly with the operating range of lift coefficient.

APPENDIX 5

Generalized Equations for Airfoil Sections

Several obvious advantages result from the use of airfoil sections which can be expressed in simple mathematical form. Such sections are always the same, can easily be checked up, the characteristics are definitely assured and the effect of any desired change in the proportions can be quickly and accurately determined. The main objection to such sections apparently is the fear that sufficient variety of combinations may not be available. However, the following assumptions as to fundamental properties and variables seem to be all that could be desired in most research work for many years to come.

Fundamental Properties

(1) Rounded nose and pointed trailing edge
(2) The cambered airfoil to be made up from a basic symmetrical section of which the center line follows the chosen camber and all ordinates remain normal to the cambered center-line
(3) No sharp break in the curvature of either the basic symmetrical section or the camber line

Variables

(1) Thickness, expressed as ratio of thickness to chord

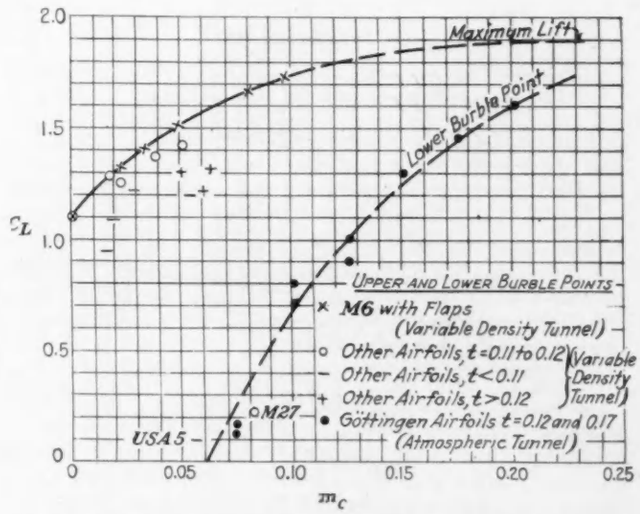


FIG. 11—CHART SHOWING THE GENERAL EFFECT OF CAMBER ON MAXIMUM LIFT AND LOWER BURBLE-POINT

m_c = (Median-Camber Depth)/Chord

(2) Camber, expressed as ratio of camber depth to chord
(3) Position of maximum section, expressed as proportion of chord from leading edge
(4) Position of maximum camber, expressed as proportion of chord from leading edge
(5) Fullness, or prismatic coefficient of the basic section
(6) Trailing-edge curvature, determined by the extent to which curvature of the camber line is maintained or reversed near the trailing edge

To meet the foregoing, working primarily from hydrodynamic considerations, as in the case of the Joukowsky profiles, or trying to deal with the different required variables by a single equation has seemed impractical. Thus the entire profile, as well as the general case of the median-camber line, is made up of separate equations for the nose and tail. The profile curves that I have found most convenient, as here given, combine in various proportions an ellipse for the nose and a hyperbola for the tail fitted by the radius of curvature as noted in Item (3) under Fundamental Properties. The same system has already been satisfactorily applied to airship hulls.

For the basic symmetrical section, the y ordinates are plotted *perpendicularly from the median-camber line*, x being measured along the curve of the camber line itself. The latter thus forms the X axis, while the perpendicular, through the maximum section, forms the Y axis. In the same way the chord of the camber line forms the X axis of the camber-line curve, while the perpendicular through the point of maximum-camber depth forms the Y axis.

The fore-and-aft dimension of the profile, taken as the curve length along the median-camber line, will be referred to as the "chord" of the profile. The chord as a base for angle measurements would be in every case the straight line joining the extremities of the median-camber line.

Following are the general equations with their derived characteristics, where a is the length *of the nose portion*, forward of the maximum section; b, the maximum ordinate of the profile, one-half thickness; d, the maximum-camber depth; e, the nose length forward of the maximum-camber depth; m, a constant depending on the position of the maximum section; n, a constant determining the fullness; p, a contant depending on the position of the maximum-camber location and q, a constant determining the amount of diminished or reversed curvature in the rear portion of the camber line.

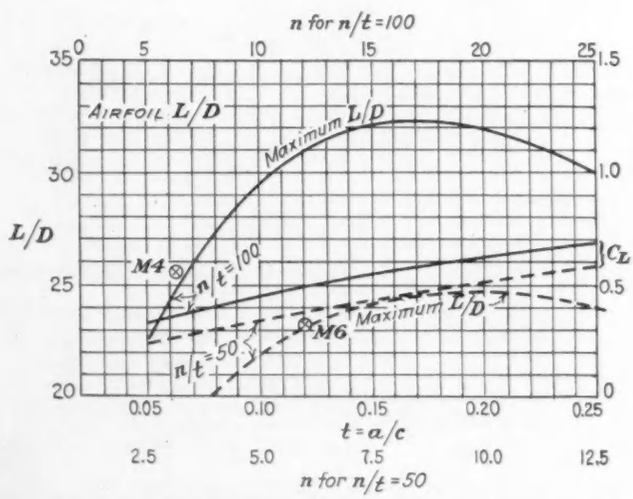


FIG. 12—LIFT-DRAG RATIO OF STRUCTURAL AIRFOILS
In This Chart n Represents the Aspect Ratio and t the Thickness Ratio

Profile Equations

$$\text{Nose.}\quad y = \pm\left(b\sqrt{1-\frac{x^2}{a^2}}\right)\cdot\left(1+n\frac{x^2}{a^2}\right)$$

$$\text{Tail.}\quad y = \pm\frac{b}{m}\left(1+m-\sqrt{1+\frac{mx^2}{a^2}}\right)\cdot\left(1+n\frac{x^2}{a^2}\right)$$

$$\text{Thickness} = 2b.$$

$$\text{Chord} = a\left(1+\sqrt{2+m}\right)$$

$$\text{Thickness/Chord} = \frac{2b}{a\,(1+\sqrt{2+m})} = t$$

$$\text{Maximum-Section Position} = \text{Nose Length/Chord} = \frac{1}{1+\sqrt{2+m}}$$

Camber-Line Equations

In these d is the maximum height of the median-camber curve and e is its distance back of the nose.

$$\text{Nose.}\quad y = d\left(1-\frac{x^2}{e^2}\right)$$

$$\text{Tail.}\quad y = \left(\frac{d}{1+p}\left[\tfrac{3}{2}+p-\sqrt{\tfrac{1}{4}+\frac{x^2}{e^2}(1+p)}\right]\right)\cdot\left(1-q\frac{x^3}{e^3}\right)$$

$$\text{Chord} = e\left(1+\sqrt{2+p}\right)$$

$$\text{Camber Ratio} = \frac{d}{\text{Chord}} = \frac{d}{e(1+\sqrt{2+p})} = m_c$$

$$\text{Maximum Camber Position} = \frac{e}{\text{Chord}} = \frac{1}{1+\sqrt{2+p}}$$

In the special case where the position of maximum camber is in the center, the tail equation given above becomes:

$$y = d\left(1-\frac{x^2}{e^2}\right)\cdot\left(1-q\frac{x^3}{e^3}\right)$$

Each of the profile equations, as will be noted, consists of two main factors. The first one in the case of the nose is a pure ellipse, and in the case of the tail a pure hyperbola. The second factor, common to both, is for the purpose of increasing the fullness characteristic above that given by the ellipse-hyperbola combination alone. For the usual case where the maximum section is 30 per cent from the leading edge, $m = 31/9$. When $m = 1$, the maximum section occurs at 36.6 per cent from the leading edge, which is probably the maximum ever desired. The practical range of n will be from 0 to about ¼. In Fig. 10 is shown a comparison of three different sections that are similar in general proportions: the Joukowsky, which has not enough fullness for much practical use; a section derived from the equations where $m = 31/9$ and $n = 0$; and the Clark Y, a symmetrical version of which can be approximated from the above equations by putting $n = 0.1$.

In the camber-line equations the nose is always a

parabola and the tail consists of two main factors; the first a hyperbola, and the second a modifying factor in case a straighter or a reflex trailing-edge is desired. Here, in the simplest case $p = m$ but can be different as in the case of the Clark Y and other well-known sections in present use, and q will probably range from 0 to about ¼. The equations may look somewhat complicated at first sight but are really very simple in practical use, as soon as the various controlling proportions are decided upon.

APPENDIX 6

Lift-Drag Ratio

In the paper itself virtually no use has been made of the still very common L/D ratio. In usual practice, dealing directly with weight and drag as separate entities is better and safer than the use of arbitrary ratios. Nevertheless, certain ideal cases exist which are of more or less academic interest as pointing the way to possible developments, in which this and similar ratios can be used to advantage. The example given below will show a legitimate use of this ratio within proper limits, at the same time indicating the futility of using it for direct purposes of practical design.

The general airfoil-drag equation as given in the paper proper is

$$C_D = C_L^2 (1 + \delta)/\pi n + (0.007 + 0.1t^2) \cdot (1 + 0.8 C_L^3)$$

Assigning to δ an average value of 0.04, using n/t as a gage of the structural requirements and dividing through by C_L we get

$$D/L = 0.33 C_L/t(n/t) + (0.007 + 0.1t^2) \cdot \left[(1/C_L) + 0.8 C_L^2\right]$$

Differentiating with respect to C_L and equating to 0, we get the following equation:

$$6.6 C_L^2/\left[t(0.07 + t^2) \cdot (n/t)\right] + 1.6 C_L^3 - 1 = 0$$

from which the value of $C_{\hat{L}}$ corresponds to the maximum L/D is determined.

Fig. 12 shows a plot of the maximum attainable L/D of simple airfoils for values of n/t of 50 and 100 respectively, also the lift coefficient at which such maximum occurs in each case. For airfoils, the n/t ratio is a more general expression serving a similar purpose to the $\hat{y}(a = d)$ ratio used for actual airplanes. Its numerical value in any particular case will, of course, depend on the structural efficiency, degree of taper and the actual magnitude of the weight, span and similar factors. Large variations in size were discussed in my paper entitled, Some Fundamental Economics of Aircraft Operation[10]. The two values here given are supposed to represent the usual extremes. Of interest in this connection is that a 5-to-1 taper permits the n/t ratio to be just about doubled, as compared with a rectangular cantilever wing, other things being equal.

[10] Presented at a meeting of the American Society of Mechanical Engineers held at Akron, Ohio, Oct. 21-23, 1929, and to be published in *Aeronautical Engineering*.

[11] See Advisory Committee for Aeronautics (British) Reports and Memoranda Nos. 1006, 1133 and 1146.

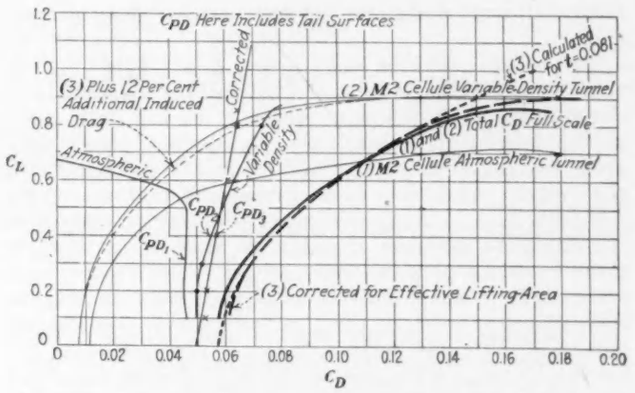


FIG. 13—DETERMINATION OF THE PARASITE-DRAG COEFFICIENT, C_{PD}, FROM WIND-TUNNEL AND FLIGHT TESTS

The curves are of particular interest as showing the rather large values of wing thickness necessary for maximum L/D when structural requirements are taken into account, and this shows how easily one can be led astray by comparing directly the L/D values of various unrelated and arbitrarily proportioned airfoils.

The n/t ratio as here used can be simply assumed but in any practical case it must be determined by methods such as outlined in the paper, and any assumption of constant n/t presupposes a range of conditions for which it is justifiable. Even so, such curves of L/D are not directly practical because the other items of drag applying to the actual airplane cause the maximum L/D of the entire airplane to be not only different but to occur at a different value of the lift coefficient than is the case for the airfoil alone, not to forget also that the effective wing-area is usually somewhat different for the complete airplane.

Thus, even at the best, considerations of airfoil L/D have very limited application, and the L/D as ordinarily used, derived from atmospheric-wind-tunnel tests on arbitrarily proportioned airfoils, is hardly worth the trouble to figure it out. That hybrid makeshift, L_{max}/D_{min}, is of still less use.

APPENDIX 7

Computation of Parasite Drag

Considerable designing apparently is still done on the basis of separating the total drag into wing drag and parasite drag. This has the disadvantage not only of lumping together very dissimilar types of drag, as brought out in the paper, but is also very commonly subject to grave errors of direct computation. Fig. 13, representing curves from an actual airplane[11] is shown as an example. The heavy full-line curve at the right, marked (1) and (2), represents the flight-tested full-scale polar. The curve marked (1) is the wing-cellule polar from an atmospheric tunnel.

The ordinary conventional method of calculating the parasite drag is simply to subtract one from the other, which gives the curve toward the left, marked C_{PD_1}. As one step toward greater accuracy, we have the upper full-line curve representing the same wing cellule, calculated from airfoil tests made in the variable-density tunnel. If these drag values are subtracted from the

same full-scale polar, we get the parasite drag marked C_{PD_2}.

The cellule curve is now subject to a further correction obtained by adding the minimum additional induced-drag of 12 per cent, which gives the dash curve at the left. This being done, an obvious discrepancy still remains because the maximum-lift coefficient for the cellule curve does not correspond to that of the full-scale polar. If, however, the effective wing-area of the complete airplane is taken at such a value as to make these maxima the same, we get the corrected full-scale curve in heavy dash lines at the right. If we take the difference between the two dash curves, we get the parasite-drag curve labeled C_{PD_3}. If now we apply the results to the drag equation given in the paper, using the actual wing proportions, we get for the complete full-scale polar the heavy dotted curve marked (3). This, as usual, fits the corrected actual curve very closely for a considerable range of values but begins to depart from it at the higher values of the lift coefficient near the burble point. The burble point here happens to be particularly low, as the *M*2 section used is a symmetrical one.

Among other things, these curves show that even such fundamental abstractions as the lift and drag coefficients, separately, should be used with great discretion and understanding of just what they mean. In most cases, the use of the lift and drag themselves is safer and more convenient.

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The Variation in Engine Power with Altitude, by W. D. Gove, National Advisory Committee for Aeronautics Report No. 295

THE DISCUSSION

CHAIRMAN C. H. CHATFIELD[12]:—Mr. Upson's paper is particularly interesting, not only from the thorough consideration he has given to the problem, but also as showing that the distinction between engineers interested in lighter-than-air work and those interested in heavier-than-air work is perhaps rather less than some of us may have thought. To see Mr. Upson pass so successfully to a field other than that to which he devoted such a large part of his time is most interesting.

EDWARD P. WARNER[13]:—I should like to sit down with a slide rule and a pencil and make some more analyses of the figures, but I can say at once that the paper gives a method of enormous value. Approaches to the fringe of this subject, so to speak, have been made from time to time, but Mr. Upson is the first person who has really taken the trouble and had the necessary knowledge and ability to work it out not merely for the wing section but for the wing as a whole, as to both aerodynamic and structural qualities. It is a method that certainly ought to be more generally applied. Mr. Upson has given us a very good start. I hope that this method will be extended to calculating the performance of the airplane, which is a step farther.

I raise one point, by way of starting an argument; that is, the necessity of being very careful with these general mathematical methods that we do not overstep the bounds of their practical application. Mr. Upson has called our attention to that danger in several cases. For instance, almost all of these purely mathematical aerodynamic analyses, which take no account of maximum lift-coefficient, go on forever. Once or twice I have been embarrassed, in working out by formula the ideal lift-coefficient for a certain condition of flight, to find that it was almost four times as high as the maximum for the wing and was really non-existent.

(*Concluded on p. 37*)

[12] M.S.A.E.—Aeronautic engineer, Pratt & Whitney Aircraft Co., Hartford, Conn.

[13] M.S.A.E.—Editor, *Aviation*, New York City.

Gum in Gasoline

By E. B. HUNN[1], H. G. M. FISCHER[2] AND A. J. BLACKWOOD[3]

PENNSYLVANIA SECTION PAPER — *Illustrated with* DRAWING, PHOTOGRAPH AND CHARTS

STATING that copper-dish gum is no index of the quantity of gum deposited in an internal-combustion engine, the authors recommend that this test be abandoned. Except for values that are identically or approximately the same as for the dissolved gum, porcelain-dish gum is also unreliable. As a substitute for these, two tests that have been developed as a result of work at the laboratories of the Standard Oil Development Co. and that seem to agree closely with actual operating-conditions are recommended. One determines dissolved or preformed gum by evaporating gasoline under conditions such that no additional gum will be formed during the test, while the other is an accelerated oxidation test for stability, in which oxidation at a temperature of 212 deg. fahr. serves as a measure of the stability of the gasoline.

To evaluate the maximum permissible gum-content that is consistent with satisfactory engine operation, tests were made in the laboratory on a single-cylinder Delco engine, upon a Buick car and a laboratory apparatus designed to imitate the conditions existing in the intake system of an engine. As a result of these tests, particularly those from the passenger-car, a dissolved-gun content of about 15 mg. of gum per 100 cc. is believed to be permissible, although further tests will be necessary to determine this point definitely.

Points brought out in the discussion include the necessity for two tests to determine the gum content, the fact that the gum as deposited on a heated metal surface would dissolve in gasoline but after coking would dissolve only in acetone or chloroform, that the dissolved-gum test should be used shortly before the fuel is put into service and not at the time of shipment from the refinery, that the only effect of gum is to deposit on the inlet valves and stems, and that the whole problem of gum formation is one that can be controlled by the refiner.

SIX TO EIGHT years ago few people had heard of gum in gasoline. Today, with the universal production of cracked gasoline of the high antiknock standard demanded by the automotive industry, the refiner has been brought face to face with the gum problem and has been diligently studying practical methods of solving it. He has been handicapped by a general misunderstanding of the true factors involved in gum formation and a lack of adequate testing-methods for gum.

The copper-dish test has for many years been universally used for determining gum in gasoline. The results of this test are extremely erratic and cannot be relied upon to predict how much gum a given gasoline will deposit in an engine. The inadequacy of this test has been pointed out by others and is shown strikingly by the data presented in the following pages.

Valuable contributions to a better understanding of the chemistry of gum formation, methods of testing for gum and its effect on engine operation have been reported by various investigators. From their results and from data collected in our laboratories during the last three years, which agree in substance, the following pertinent conclusions with respect to gum in gasoline have been drawn:

(1) Cracked gasoline, as it comes from the stills, is free from actual gum. Usually, however, it contains certain unstable hydrocarbons that will form gum during storage.

(2) Gum formation from these unstable constituents in cracked gasoline is essentially an oxidation process, the gum containing a high percentage of combined oxygen. Also, tests have shown conclusively that gum will not be formed in the entire absence of oxygen.

(3) The gum formed by oxidation during storage remains dissolved in the gasoline except in extreme cases. The quantity of this dissolved or preformed gum is what chiefly concerns us with respect to engine operation. This dissolved gum is sprayed from the carbureter with the gasoline and deposits on the heated portions of the intake manifold and on the inlet valves. Imperfect engine-operation and loss of power may result either from a partial clogging of the manifold or delayed seating of the inlet valves. In severe cases, engine failure may be caused by complete sticking of an inlet valve. No other deleterious effects on the engine have been noted. Only a fraction of the dissolved gum in the gasoline is deposited in the intake system, the percentage of the total depending on the design and operating conditions of the engine. The undeposited gum is carried into the combustion-chamber and burned with the charge.

(4) The dissolved or preformed gum is the only criterion with respect to the gasoline which determines the quantity of deposit that will be formed in the intake system. Unstable constituents, or potential gum, in the gasoline are not transformed perceptibly into gum deposits during the passage of the air-fuel mixture through the manifold. Hence a very unstable gasoline, one that would contain a prohibitive gum-content after storage, would give satisfactory engine operation if consumed immediately after manufacture.

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[2] Standard Oil Co. of New Jersey, Elizabeth, N. J.

[3] Jun. S.A.E.—In charge of lubrication testing laboratory, Standard Oil Development Co., Elizabeth, N. J.

(5) At present we have no authentic data from which the permissible dissolved-gum content for satisfactory engine operation can be defined. Our tests with a Buick car indicated that a 10-mg. gum-content per 100 cc. of gasoline is safe, whereas a 25-mg. gum-content gave definite signs of trouble in 5000 miles of operation. Obviously, the correct answer to this question depends on reliable data from many more tests on engines of various makes operating under normal conditions.

(6) A gasoline that is to be marketed must be sufficiently stable toward oxidation to contain not more than the permissible dissolved-gum content at the time of consumption. Since the intervening storage period is very indefinite, an adequate safety factor must be allowed so as to assure the product being satisfactory to the consumer, even after the most severe oxidizing conditions to which it will be subjected during storage. The many factors affecting gum formation during storage are of interest chiefly to the refiner and will not be considered here.

Significant Gum Tests

Voorhees and Eisinger, in their paper entitled, The Effect of Gum in Gasoline[4], which was presented at the 1929 Annual Meeting of the Society, and the Fourth, Fifth and Sixth Annual Reports of the Joint Benzol Research Committee of the University of Leeds and the National Benzol Association of Great Britain have shown that only two gum tests are of practical significance for gasoline. These are (*a*) a method of determining dissolved or preformed gum that will indicate whether the gasoline is suitable for use in an engine at or near the time of testing and (*b*) an accelerated oxidation or stability test that will assure the product being sufficiently stable to remain free from excessive gum during storage.

The copper-dish test fulfills neither of the foregoing requirements. Statements have been made that a gasoline showing low copper-dish gum, 10 to 15 mg. per 100 cc., at the time of manufacture is certain to be stable during storage, which apparently is the only justification for keeping this test as a basis for a gum specification. We have found that many cracked gasolines with low initial copper-dish gum may be very unstable in storage. This is illustrated strikingly by the data presented in Table 1 on two samples made by cracking Mid-Continent gas-oil at 650 lb. pressure.

Since oxidation and gum formation during evaporation from a copper dish depend largely upon the volatility of the sample, a gasoline such as *B* in Table 1 would clearly have much less chance than *A* to form gum during this test, whereas in storage both *A* and *B* would be exposed to equal oxidizing conditions. Sample *B* actually is more unstable than *A*, yet in evaporating from a copper dish the lower-boiling unstable fractions in *B* volatilize too rapidly to be oxidized to gum.

This indicates clearly that the copper-dish test may be expected to give very misleading conclusions as to the true stability or gumming-tendency of a gasoline, particularly if it is of high volatility, such as *B*, or has unstable constituents predominating in the lower or middle-boiling fractions.

Air-Jet Dissolved-Gum Test

Dissolved or preformed gum is determined by evaporating the gasoline under conditions such that no additional gum will be formed during the test. The Bureau of Mines steam-oven test[5] accomplishes this by evaporating in the absence of air and is in use either as originally proposed or in modified form by various laboratories.

For the last year a dissolved-gum method developed in our laboratories has been found to give results agreeing closely with the Bureau of Mines method, and, in addition, we believe has certain advantages over the latter as a routine test. The procedure consists in directing a jet of air vertically toward the center of a 100-cc. sample of gasoline contained in a glazed porcelain dish on a steam bath, using as much air as possible without splashing the gasoline over the rim of the dish. A suitable jet consists of a tube having a ½-in. inside diameter loosely packed with cotton, the open end being covered by a fine-mesh wire screen. Complete evaporation of a 400-deg. fahr. end-point gasoline requires from 12 to 15 min. The residue is dried at 100 to 105 deg. cent. (212 to 221 deg. fahr.) to constant weight, which requires from 2 to 3 hr. for moderate gums. Smaller samples, 20 cc., and smaller dishes can be used with equally good results, the gum being calculated to milligrams per 100 cc.

This method has been checked by replacing the air-stream with both nitrogen and carbon dioxide, indicating that no perceptible increase in gum due to oxidation by the air occurs. Obviously, the air jet hastens evaporation and lowers the temperature of the gasoline, thereby preventing the oxidation that may occur during normal evaporation of gasoline from an open porcelain dish.

Accelerated Oxidation Test for Stability

Methods have been described for measuring the stability of gasoline by an oxidation test at elevated temperature[6]. For more than two years we have been experimenting with similar controlled oxidation tests that consist of heating the gasoline in a bomb along with air or oxygen for definite periods of time and determining the gum in the oxidized sample.

Our recent modification of such an oxidation test incorporates a recording pressure-gage, *a*, Fig. 1, for following the course of oxidation and resulting gum formation of the sample in the bomb. In performing this test, an open 8-oz. bottle containing the sample is placed in the bomb *b* and the cap of the latter screwed on tight. Next, the small-diameter copper tubing *c* leading to the pressure gage is connected to the bomb

TABLE 1—GUM-FORMATION DATA ON TWO CRACKED GASOLINES

Sample	*A*	*B*
Initial Boiling-Point, deg. fahr.	86	84
Amount Distilled		
At 212 Deg. Fahr., per cent	25.0	44.0
At 302 Deg. Fahr., per cent	51.0	85.8
Final Boiling-Point, deg. fahr.	444	354
Initial Copper-Dish Gum, mg. per 100 cc.	348	13
Porcelain-Dish Gum after 2 Months' Storage		
In Dark, mg. per 100 cc.	28	600
In Light, mg. per 100 cc.	687	2,435

[4] See S.A.E. JOURNAL, June, 1929, p. 584.

[5] See Bureau of Mines Report, Serial No. 2686.

[6] See S.A.E. JOURNAL, June, 1929, p. 584; also the Fourth, Fifth and Sixth Annual Reports of the Joint Benzol Research Committee of the University of Leeds and the National Benzol Association of Great Britain.

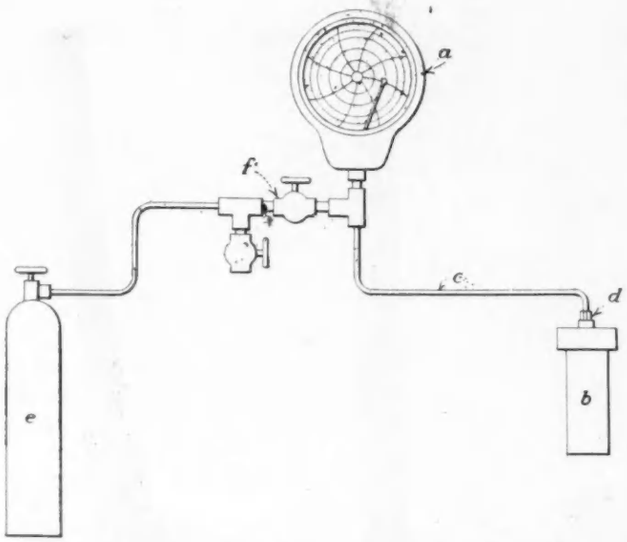

FIG. 1—APPARATUS FOR MEASURING STABILITY OF CRACKED GASOLINES TOWARD OXYGEN

In Making the Accelerated Oxidation Test for Gum Formation, an Open 8-Oz. Bottle Containing the Sample Is Placed in the Bomb *b*, the Cover of Which Is Screwed on Tightly, and Oxygen Is Admitted under Pressure from the Cylinder *e* Until the Pressure on the Recording Gage *a* Registers 100 Lb., When the Bomb Is Heated to 212 Deg. Fahr. by a Steam Bath. The Pressure Then Rises to a Maximum That Remains Practically Constant Until Oxidation Starts and the Gasoline Breaks Down, This Point Being Indicated by a Sharp Break in the Pressure Curve Drawn by the Gage

by a compression coupling *d*. Oxygen from the cylinder *e* is admitted to the bomb until the gage reads 100 lb. The oxygen supply is then cut off by closing the needle-valve *f* and the bomb heated at 212 deg. fahr. by a steam bath.

After immersing the bomb in the heating bath, the pressure rises to a maximum which remains practically constant, in the absence of leaks, until oxidation starts and the gasoline breaks down. This is indicated by a sharp break in the curve and a rapid drop in pressure. The length of time before the breakdown, the so-called induction period, is taken as a measure of the stability of the sample. Typical oxidation-stability curves are shown in Fig. 2. The curve for the vapor-phase gasoline indicates that it is definitely unstable; cracked gasoline No. 1 is somewhat more stable; whereas cracked gasoline No. 2 is sufficiently stable for prolonged storage.

Correlation of the results of the above test and ordinary storage has not progressed far enough to state definitely the permissible minimum induction-period necessary to assure the product being satisfactory for storage. However, indications are that a gasoline which stands up for 1½ to 2 hr. under this test is sufficiently stable for usual storage conditions.

The tests include series on single-cylinder Delco engines, a Buick car and a laboratory apparatus designed by engineers of the Ethyl Gasoline Corp. to imitate the conditions existing in the intake system of an engine. Their object was to determine more definitely which of the laboratory testing methods is significant with respect to gum deposition in an engine and to define, if possible, the permissible maximum gum-content for satisfactory engine operation.

Tests Made with Engines

The engines used were two water-cooled Delcos, each fitted with a special intake manifold, 5 in. long, horizontally placed and electrically heated. A thermocouple at the engine end of the manifold recorded the temperature of the air-fuel mixture. Fixed jets were used in the carbureters. Twenty samples were run, representing highly cracked gasolines of good antiknock value containing varying quantities of gum. The conditions chosen were as follows: gasoline per test, 25 gal.; engine speed, 1200 r.p.m.; water-outlet temperature, 180 deg. fahr.; mixture temperature, 170 deg. fahr.; throttle opening, maximum possible with the available loading resistances.

Under these conditions we found that engine No. 1 required 100 hr. per run, whereas engine No. 2 required 120 to 130 hr., the variation being due to the difference in the available methods of applying the loads. Also, the head of engine No. 2 was of different construction, with a larger water pocket surrounding the inlet valve. These differences resulted in heavier gum deposits in engine No. 2 than in No. 1, making a direct comparison of the two gasolines impossible unless they both were run on the same engine.

At the completion of each run the engine head and inlet-valve stem were photographed and the gum and carbon deposits recovered and weighed. Since photographing the interior of the heated inlet manifold, where from 70 to 90 per cent of the gum was deposited, was impossible, the pictures of the valves alone are not conclusive. As a matter of interest only, a few are shown in Fig. 3.

The significant results from the 20 runs made on

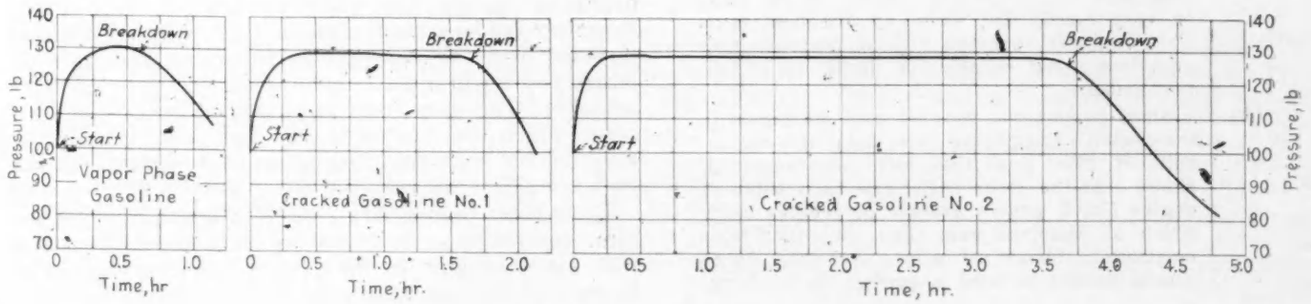

FIG. 2—TYPICAL OXIDATION STABILITY CURVES

These Show the Definite Instability of the Vapor-Phase Gasoline, the Slightly Greater Stability of Cracked Gasoline No. 1, and the Marked Stability of Cracked Gasoline No. 2 Which Can Be Stored for Prolonged Periods

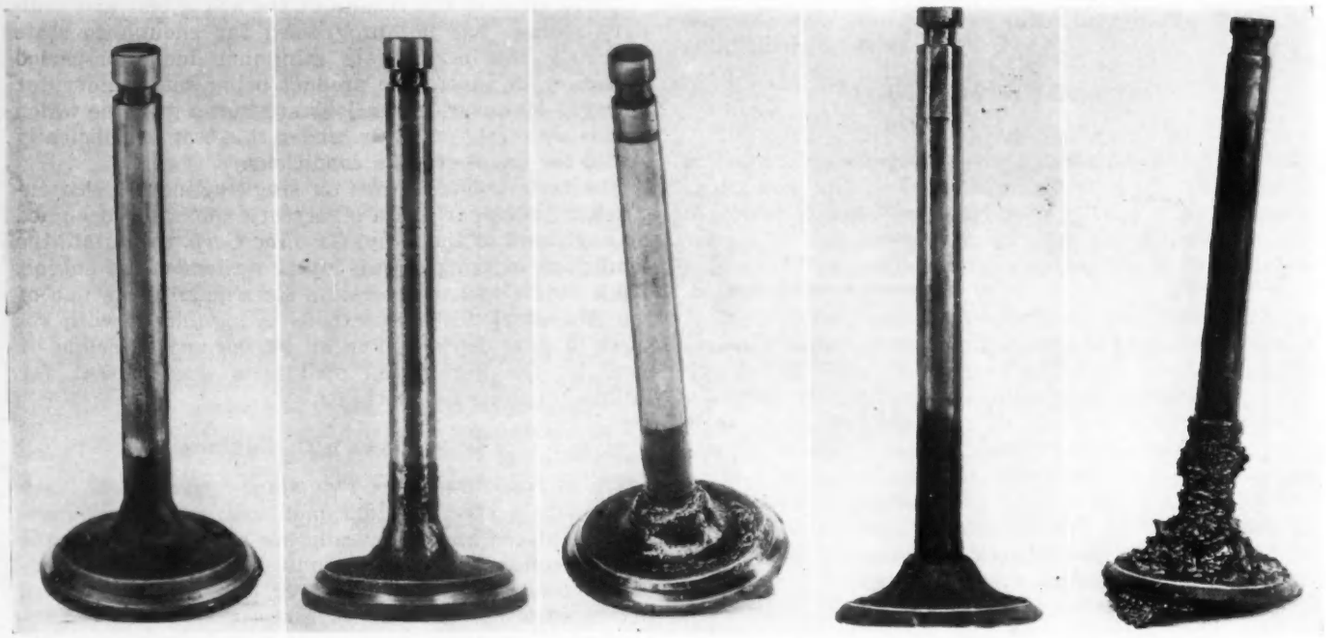

FIG. 3—SOME TYPICAL VALVES FROM THE DELCO ENGINES USED IN TESTS FOR GUM DEPOSITION

The First Three Valves from the Left Were from Engine No. 1 and the Other Two from Engine No. 2. The Sample Numbers Correspond to Those in Table 2, Which Gives Particulars of the Gasolines Tested and the Results Obtained

Sample No.	5	9	1	17	16
Copper-Dish Gum, mg.	258	320	180	9	493
Porcelain-Dish Gum, mg.	19	20	17	2	232
Dissolved Gum, mg.	8	14	16	1	51

these Delco engines are given in Table 2. The gum results shown for each sample are the averages of tests by three laboratories for copper-dish, porcelain-dish and dissolved gum. Samples were tested at the beginning of each run and again near the end to make sure that additional gum had not formed during the progress of the engine test.

Conclusions Deduced from Tests

From these tests it is concluded that

(1) *Copper-dish gums* are no indication of the quantity of gum deposit formed in the engine. Samples such as Nos. 3, 5 and 9 with copper-dish gums between 200 and 300 mg. gave roughly the same deposit as did samples Nos. 10*B* and 12, both of low copper-dish gum-content.

(2) *Porcelain-dish gums* also do not check with engine deposits, except where they are the same or very nearly the same as dissolved gum. This indicates that the regular porcelain-dish gum test is not reliable for predicting engine performance.

(3) *Dissolved gums* agree roughly with deposition in the engine. Comparing these data with the results of other tests that have been reported[7] shows that the Delco tests, especially those of engine No. 2, gave a greater percentage deposition of dissolved gum than the other tests mentioned. Hence we believe that these Delco results cannot be used to predict the performance of the same or similar fuels in automobile engines. (See Fig. 4.) Their value lies in the correlation we have been able to get between gum deposition and laboratory tests, showing most conclusively which of the latter are significant.

(4) *Carbon deposits* seem to bear no relation to the quantity of gum in the fuel.

(5) *Sticking valves* were not common during these tests. Engine No. 1 gave no trouble from this source. Engine No. 2 developed a sticking inlet valve on samples Nos. 4 and 9*A*, both with high dissolved-gum, which were the only serious cases.

Ethyl Gasoline Corp. Gum Apparatus

These results are included with the engine tests since the apparatus was designed to approximate conditions existing in the inlet ports of an engine, where gum is deposited. Briefly, this apparatus consists of a Holley vaporizer fed by gravity from a 1-qt. tank, the air-fuel mixture from the vaporizer being drawn by a vacuum pump over an iron plate maintained at a constant temperature. The gum from the mixture deposits on this plate, which was heated to 400 deg. fahr. and can be removed for weighing. One quart of fuel was used for each test, the time required being about 1 hr.

From these preliminary results, given in Table 3, the same conclusion is indicated as from the Delco tests, that dissolved gum in the fuel is the only significant figure and that potential gum is of no importance with respect to the quantity of deposit formed. The last two columns of the table show that virtually all of the dissolved gum in the sample is deposited in this apparatus,

[7] See S.A.E. JOURNAL, June, 1929, p. 584; also the Fourth, Fifth and Sixth Annual Reports of the Joint Benzol Research Committee of the University of Leeds and the National Benzol Association of Great Britain.

TABLE 2—DELCO-ENGINE TEST DATA

Sample	Description	Gum Tests on Sample, Mg. Copper Dish	Porcelain Dish	Dissolved	Total Gum-Deposit, Mg. Per 100 Cc.	Per Gal.
	Engine No. 1					
1	Antiknock Gasoline plus 1 Per Cent of High-Gum Gasoline	180	17	16	4.4	168
3	2 Parts of Antiknock Naphtha plus 1 Part of Vapor-Phase Gasoline	212	13	17	1.5	56
5	Acid - Treated Vapor - Phase Gasoline	258	19	8	0.3	12
7	Antiknock Gasoline plus Copper Oleate	277	130	19	3.3	124
9	Untreated Cracked Gasoline	320	20	14	1.7	64
10	Antiknock Naphtha	29	5	5	0.5	20
10*A*	Antiknock Naphtha plus 2 Cc. of Tetraethyl Lead	46	15	6	0.4	16
10*B*	Antiknock Naphtha Blend	18	6	6	1.6	60
13	California Gasoline	15	5	3	0.4	16
14	Antiknock Gasoline plus 5 Per Cent of Raw Vapor-Phase Gasoline	172	19	19	4.9	184
	Engine No. 2					
1	Antiknock Naphtha plus 1 Per Cent of High-Gum Gasoline	180	17	16	6.5	248
4	Untreated Vapor-Phase Gasoline	243	968	122	29.0	760
6	Antiknock Naphtha plus Copper Oleate	240	126	11	5.8	220
8	Antiknock Naphtha plus 1 Cc. of Tetraethyl Lead	15	4	3	0.7	28
9*A*	Untreated Cracked Gasoline	365	50	36	8.0	304
11	Antiknock Naphtha plus 1 Cc. of Tetraethyl Lead	12	3	3	0.8	32
12	Vapor-Phase Gasoline after 2½ Months' Storage	170	33	8	3.8	144
15	Vapor-Phase Gasoline Blend after 4 Months' Storage	17	10	6	3.1	116
16	Acid - Treated Vapor - Phase Gasoline after 5 Months' Storage	493	232	51	1.0	72[a]
17	Acid - Treated Vapor - Phase Gasoline	9	2	1	0.4	16[a]

[a] Inlet-manifold heater off during these runs.

as contrasted with a recovery of roughly from 10 to 25 per cent when the same fuel is run in a Delco engine.

Buick Test-Car Data

Three tests were made with a Buick Master-Six automobile to define more clearly the maximum permissible quantity of gum for satisfactory engine operation. The fuels were prepared by blending a high-gum gasoline with a good grade of motor gasoline to give a 10, a 25 and a 50-mg. dissolved-gum content. The significant data are shown in Table 4. Test No. 7, 10-mg. gum,

TABLE 3—ETHYL GASOLINE CORP. GUM-APPARATUS DATA

Sample No.	Description	Gum, Mg. per 100 Cc. Copper Dish	Porcelain Dish	Dissolved	Deposition on Plate
1	Finished Gasoline *C*	..	5	4	4
2	California Gasoline	14	5	2	2
3	Finished Gasoline *A*	14	3	3	0.6
4	Finished Gasoline *B*	25	14	8	5
5	Columbia Gasoline after 6 Months' Storage	131	20	13	9
6	Vapor-Phase Gasoline after 5 Months' Storage	508	240	49	39
7	Finished Gasoline *F* Blend	10	4	2	2
8	Finished Gasoline *E*	162	86	33	34
9	Finished Gasoline *D*	13	5	4	2
10	Columbia Gasoline after 4 Months' Storage	104	16	10	7

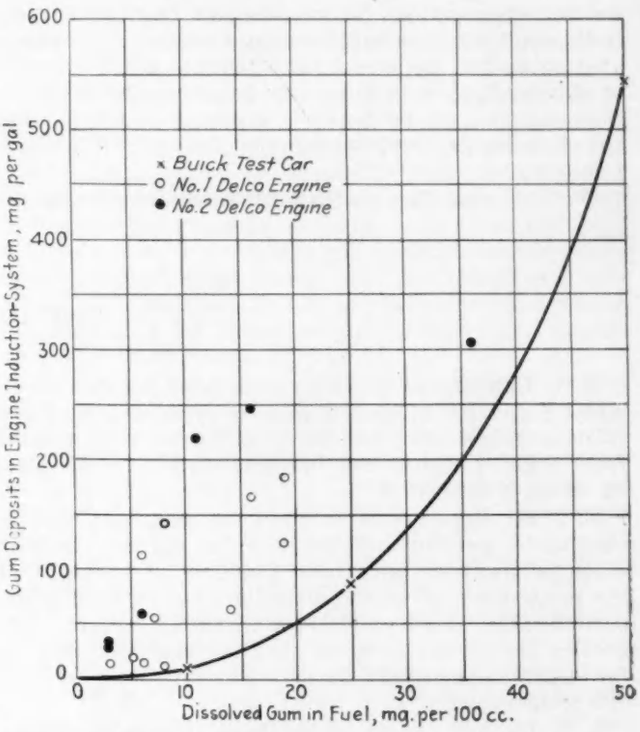

FIG. 4—ENGINE GUM-DEPOSITS PLOTTED AS A FUNCTION OF DISSOLVED GUM

The Data from Which the Various Points Were Plotted Were Obtained from the Buick-Car and Delco-Engine Tests. Calculations Based on the Curve Show That a Fuel Containing 15 Mg. of Gum per 100 Cc. Would Leave a Deposit of 17 Gm. after 10,000 Miles of Operation

showed perfect operation throughout the 5000 miles. In test No. 8, 25-mg. gum, the loss of power after 3000 miles was perceptible but no valve trouble occurred, whereas in test No. 6, 50-mg. gum, the loss of power was very marked and slight trouble from sticking inlet valves was experienced.

From these data the curve of Fig. 4 was made, plotting the milligrams of deposit per gallon of fuel used against the dissolved-gum value for the fuel. From this curve it is calculated that a fuel containing 15 mg. of gum per 100 cc. would leave approximately 17 mg. of deposit after 10,000 miles of operation of this particular car. This suggests that a 15-mg. dissolved-gum content is permissible, although further tests on multiple-cylinder engines will be made to substantiate this tentative conclusion.

Conclusion

As a result of our investigations and in the light of considerable published material on the subject, we recommend that the copper-dish gum-test be abandoned in

TABLE 4—BUICK TEST-CAR DATA

Test No.	7	8	6
Gum in Fuel			
Dissolved, mg.	10	25	50
Copper Dish, mg.	..	85	89
Distance Run, miles	5,000	5,006	2,722
Fuel Consumption, gal.	315	332	203
Gum Deposits Recovered			
From Heated Riser and Manifold, gm.	0	19	50
From Inlet Valves and Ports, gm.	3.6	9	60
Total, gm.	3.6	28	110
Deposit per Gallon of Fuel, mg.	11	88	543

writing specifications for gasoline and that two suitable tests, similar to our rapid-evaporation test and accelerated aging-test, be adopted and that the whole question of standardization of gum tests be studied by the Fuels Subcommittee of the Society's Research Committee and the Lubricants Division of the Society's Standards Committee.

We feel that the question of which gum tests are significant has been amply demonstrated and that the most important question still unanswered concerns the permissible maximum gum-content for satisfactory engine operation. When the refiner has the answer to that, he will have a target to aim at and can more intelligently work out the correlation between laboratory tests for stability and actual gum formation during storage.

THE DISCUSSION

H. T. KENNEDY[8]:—In these tests using the thin sheet-metal plates on which the gum is measured, was any relation found between the quantity of gum formed from a given sample and the temperature of the plate on which it is formed?

E. B. HUNN:—So far we have not made any tests at varying temperatures of the plate, but the reports from other laboratories have been that the temperature of the metal does affect the quantity of gum that is deposited. The higher the temperature, usually, the greater percentage of gum that is deposited. But so far with this laboratory device we have operated at only one temperature.

J. B. RATHER[9]:—One of the most interesting things about this gum paper is the frankness with which the problem has been approached, and it is a very wise change from the point of view of the oil companies of a number of years ago. My organization was among the earliest, if not the earliest, to study gum. At least four years ago we went through a very extensive series of experiments correlating engine performance with laboratory tests and correlating the storage conditions and laboratoy tests. This work confirms that of the Standard Oil Co. of New Jersey, and, while we agree that the copper-dish test is misleading in a number of ways, I think we can be sure of one point in respect to the copper-dish test; that is, a gasoline which will show no gum on the copper-dish test will give no trouble in an engine.

On the change from potential to actual gum, in storage, we made a series of tests extending over six months or more in containers varying from a 4-oz. bottle to 10,000 gal. on a gasoline containing approximately 600 mg. of potential gum and practically no actual preformed gum. We found that, in containers larger than a barrel, the change in gum, that is, the instability of the gasoline, was relatively small. We could not measure it all in the larger containers, but we found with containers of 5 and 10 gal. and less, that the smaller the container is, the greater is the tendency to change. Our conclusion was that catalytic influences, that is, oxidation accelerated by various catalytic agencies, like the metal and things of that sort, were the prime factors in changing this so-called potential gum into the actual gum that causes trouble. From our point of view, the problem is not one of storage in large quantities but of storage in small quantities, such as are found in the vacuum tank of an automobile or in a carbureter.

Two Tests for Gum Needed

J. B. HILL[10]:—All of us who have been working on gum will agree with Mr. Hunn that we need two tests; we need something for this potential gum and something for the dissolved gum. The tendency is to feel that only one test is necessary but that the copper-dish test is not the one, and that a test which shows only dissolved gum is considerably better than the copper-dish test. I do not altogether share Mr. Hunn's pessimism in his complete condemnation of the copper-dish test. If we are to use only one test, I think that in spite of its erratic results the copper-dish test probably gives us as good a picture as we can get and is at least as good as any test we have. As Mr. Rathers says, if a gasoline is good by that test we know that it is good.

We petroleum people ought not to frighten the automotive people unduly on the subject of gum. We are very fully alive to the situation; as a matter of fact, we are if anything a little too worked up about it. We are all working on it, we are all faced with the problem of bringing down the gum values of the high-gum high-antiknock gasoline we are producing. Very, very few commercial gasolines on the market today offer any danger of gum.

FERDINAND JEHLE[11]:—Just what is the nature of the trouble that you experience with this gum? Presumably, it is sticky inlet valves, but is it not possible that these deposits of gum may also carry through to the exhaust-valve stems?

MR. HUNN:—So far as we have been able to determine, no bad effect is experienced from the gum except on the inlet system. Gum will build up either on the heated portions in the manifold and eventually clog it or else on the inlet-valve stems only and cause them to seat imperfectly or stick.

Use of Dissolved-Gum Test

A. L. BEALL[12]:—The proposed dissolved-gum test is pertinent only for the period immediately preceding the use of the fuel. It cannot be applied at the time of shipment from the refinery to determine effectively how the fuel will act after a reasonable storage period. Thus the test might best be applied shortly before the fuel is actually put into service, and the whole gum test as proposed can be useful only when data have been accumulated on the accelerated oxidation test that is also proposed.

The paper carries the recommendation that the So-

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[10] M.S.A.E.—Chief chemist, Atlantic Refining Co., Philadelphia.

[11] M.S.A.E.—Research engineer, White Motor Co., Cleveland.

[12] M.S.A.E.—Engineer, Vacuum Oil Co., New York City.

ciety take cognizance of the gum situation and do some work on it. I would be inclined to endorse including the gum problem in the Cooperative Fuel Research, with the idea that, since it has had this much publicity, why not carry it out and finish it in the open as it has been brought out here today.

Is the air-fuel ratio in the mixture that is passed over the hot-spot in the ethyl gasoline apparatus fixed?

MR. HUNN:—The apparatus was designed to give a fixed air-fuel ratio, but of course the ratio will be different depending on the gasoline. It was designed to be as simple as possible and free from variables, to be a simple testing device for determining the quantity of gum deposited.

MR. BEALL:—Would 15 to 1 be considered ideal, if that is determined by simply using the air as a means of conveyance of the fuel vapor?

CHAIRMAN J. P. STEWART[13]:—It is within the range of working engine ratios.

Nature and Solubility of Gum

B. P. SPARKS[14]:—Can you describe the nature of this gum and tell how we can identify it when we see it and what might dissolve it? I want to identify this with gum that has been found through the fuel system and traced to flux used in soldering gasoline tanks. Working on that theory and changing the gasoline tanks eliminated the trouble from gum throughout the gasoline system, particularly in the fuel-pumps and also in crankcase-oil dilution.

MR. HUNN:—The gum, as it is deposited on a heated metal surface and before it is baked to a fairly hard residue, is soluble in gasoline. After it is baked down, it will dissolve very readily in acetone or chloroform but not in gasoline. For instance, in the evaporation test for gum the residues are not soluble in gasoline because, during the drying of the gum in the oven, it is oxidized to that insoluble state.

MR. SPARKS:—Would you describe this gum as a varnish-like substance?

MR. HUNN:—Yes, it resembles varnish somewhat.

R. T. HASLAM[15]:—Mr. Hunn has some data showing the effect of temperature on an accelerated gum-test. If I remember the data correctly, they show the same condition in the formation of gums at temperatures ranging from 100 to 210 deg. fahr. or thereabouts, indicating that the gum formed in the accelerated test at 210 deg. is of the same general character as the gum formed at room temperatures.

Throughout the solution of the gum problem we have faced the necessity of getting an adequate test for gasoline. The copper-dish test is not fulfilling the requirements. As Mr. Rather said, we may have a perfectly good gasoline with a high copper-dish gum, yet it would not pass another specification. If the copper-dish test is low, we do have usually, but not always, a good gasoline. If the copper-dish gum test is high, we still may have a good all-round gasoline.

Storage in small tanks is not as important as storage in the distributing tanks of the oil companies, on account of the short time the gasoline is in the former. It is not in the vacuum tank very long, and the gum formation is the effect not only of temperature but also of time. Formation of gum in these tanks, as was brought up later, might be due to the soldering flux accelerating the gum formation of the gasoline used in those particular cars.

MR. KENNEDY:—What effect does a very small percentage, say a fraction of 1 per cent, of heavy ends of oil in the gasoline have on the heated-plate test?

MR. HUNN:—That would also show as a residue on the gum test, unless the residue is dried in the oven for sufficient time to give it constant weight. In some cases in which gasoline was contaminated with fairly heavy oil, 3 or 4 hr. and sometimes longer was required to evaporate all of the oil.

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[15] Manager, development department, Standard Oil Development Co., Elizabeth, N. J.

Wings—a Coordinated System of Basic Design

(*Concluded from p.* 30)

A number of Continental writers have treated the general theory of airplane performance and equilibrium extensively. Their works have often suffered seriously in value because they made assumptions that were unduly general, forgetting that the assumptions had limits of application. Mr. Upson has warned us again in several cases as to that, particularly in speaking of the probable necessity of including some additional torque bracing in the wing in very thin sections. I wonder if a warning against the lower end of that curve of effective wing-taper should not be given? As a practical matter I would be interested in having Mr. Upson say if he thinks it possible to taper the wing down to a point that would be about a 4-to-1 ratio, without getting the ribs, and probably also the spar webs, substantially heavier than the pure theory, as here developed, would indicate.

RALPH H. UPSON:—Mr. Warner's question about the practical limitation of taper can be answered, I think, in this way: I have tried to take a conservative weight-variation with respect to taper variation, and have done so intentionally because the results *did* seem to be remarkably favorable to the large-degree tapers and I did not want to emphasize it any more than necessary.

By referring to the weight formula it will be seen that I have assumed the spar flanges, for example, to be proportional in weight per foot to the root weight per foot for all degrees of taper, which ignores the more favorable *curve* of stress in the case of the spar for tapered wing. I have also, similarly, taken the web weight by total load and area rather than by calculating the extreme theoretical advantage that might be derived from the very light stresses at the end of a highly tapered wing. The rib weights are more likely to run over the formula for extreme tapers, especially in large sizes where beam loads in the ribs must be considered.

With regard to the applicability of theory in general, I need perhaps only remark that no theory is ever complete, and an incompetent engineer using this or any other theory is still an incompetent engineer.

Demands of Air Passengers Challenge Engine-Builders' Ability

By Robert Insley[1]

Washington Section Paper

NOW that the speculative promotional period of aviation, in which money for experimental development was plentiful, is over, the author asserts that the aeronautic industry faces the necessity of determining what the trend of development must be and of laying plans in accordance therewith. Present conditions affect the engine manufacturer more seriously than the airplane builder, and it seems to be necessary for him to anticipate well the course of events.

Reasons are given for the fact that private citizens are not buying airplanes in any considerable number for private use, although they are eager to travel in those owned and operated by others. For the immediate future, the author foresees that the market will be for airplanes that correspond to the chauffeur-driven motor-vehicle. And, as the majority of the airplanes will be sold for carrying passengers for hire, the passengers will dictate the qualities they want in the airplanes. These the author lists as (*a*) speed, (*b*) safety, (*c*) comfort, and (*d*) economy; their order of importance depending upon the individual passenger.

Each of these properties is analyzed, and the responsibility of providing it is placed upon the shoulders respectively of the airplane operator, airplane builder and engine manufacturer. Thus, the selection of power rests primarily with the airplane constuctor, but ultimately with the operator. The aerodynamic efficiency is in the hands of the airplane designer, but the propulsive efficiency, safety from fire hazard and reduction of noise and vibration are the concern of the engine producer.

Summarizing, the author indicates courses of procedure that the requirements of air passengers seem to propose to the engine makers. The task of meeting the demands is not easy, and when they have been met more will have been discovered.

DURING the aircraft boom period we grew accustomed to prodigality in development expenditures. To attempt to guess the trend of the industry seemed futile, and all but the wisest or the poorest adopted a shot-gun attack which could hardly fail to hit something. Money was plentiful and the cost incidental. But with the industry now on a strictly business basis, we must make our shots count and must see clearly what we are aiming at before we shoot. It behooves us to determine just what the natural trend of the industry must be and to lay our plans accordingly. The situation affects the engine manufacturer even more seriously than the airplane manufacturer, because his investment in tools and equipment is considerably greater, his development period somewhat longer, and the cost of unsuccessful experiments correspondingly larger. It seems to be necessary for the engine manufacturer to indulge in the dangerous occupation of prophesying and to do it well if he hopes to avoid the expense of wild shots. His task is by no means an easy one but some tendencies by which he may at least take a sight are apparent.

The various classes of air vehicles may be divided roughly into two groups corresponding to the private and commercial motor-cars, or more exactly to the owner-driven and chauffeur-driven cars. After deciding upon the probable destinies of these two main groups, we may proceed to break down the interesting portions into subordinate groups and perhaps form an opinion about the probable engine requirements of the industry.

The owner-driven motor-car represents the bulk of the profitable motor-car business. The chauffeur-driven motor-car—including the motorcoach, motor-truck taxicab, light commercial vehicle and private car—represents a less profitable but probably less fickle market. We look forward to the day when the motor-car owner will buy an airplane instead of his second or third motor-car and so establish the owner-piloted airplane as the profitable portion of the airplane business. But at present it must be admitted that, despite the development of scores of small airplanes for private ownership and scores of light engines for use in such airplanes, the immediate aircraft market is the market corresponding to the chauffeur-driven motor-vehicle.

Where the Immediate Market Lies

Mr. and Mrs. Average Citizen are not ready to buy airplanes. Their reasons are many: first cost, maintenance, distance to airports, cost of learning to fly, safety in the air, doubtful utility, and a dozen others. Each person has his own relative order of importance of these reasons but each has a sufficient number of valid reasons to justify, in his own mind, his being without an airplane. The problem is much the same as that of owning a motorboat or a horse, except that the possibilities of the airplane are vastly greater. Whatever the reasons may be, private citizens are not buying airplanes in any noticeable number. No doubt can be entertained that they will do so in time, but to bring this about will require the combined results of the Guggenheim Aircraft Safety competition, the best efforts of production engineers, cheaper flying courses, better and more numerous airports, a considerable amount of publicity and education, and vastly more intelligent and sympathetic cooperation on the part of the newspapers in reporting accidents.

Most persons are willing, even eager, to fly and will

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pay well for the privilege, but they insist on flying in an airplane bought and maintained by someone else. Therefore the industry's immediate market, and that for several years to come, is the market corresponding to that for the chauffeur-driven motor-car. This conclusion, if it be true, does not mean that no small airplanes or engines will be sold next year. A market will always exist for small airplanes. But it does mean that the stable market for the next several years will be in the class of airplanes suitable to be flown for hire. In support of this conclusion I submit that virtually all air-school students, when questioned about their intentions, admit that they intend either to buy an airplane and carry passengers or work for someone else.

Since the majority of the airplanes will be sold for passenger carrying, the passenger carried will, in the long run, indicate to the operator what he desires in his airplane and will insist upon getting it. Air passengers in the next few years will fly in airplanes of all kinds, from the two-place open-cockpit sport planes to the 50-passenger or larger multi-engine air-transport. The engines used will vary from less than 100 to 600 or 700 hp. But, in general, regardless of the size of the airplane and engine used, the desires of the passenger will be similar. The air passenger has four general requirements: speed, safety, comfort and economy; their order of importance depending upon the temperament of the individual customer. There may be minor requirements not associated directly with these, but, in general, these four represent the important problems confronting the air-transport operator.

Engine Types for Higher Speed

The speed requirement never will be satisfied. Speed is important, otherwise the public would have no excuse for air travel, but for each class of air travel there is an economical speed limit beyond which we cannot afford to trespass. Speed, directly and indirectly, costs money, and the transport operator himself must decide how much speed he can afford to furnish and must power his airplane accordingly. However, several improvements can be made by the engine manufacturer to better the speed without increasing the size or reducing the economy of the engine, provided the airplane builder is satisfied with the effect of such changes on the cost of the engine. Speed can be increased by increasing the power or by improving the propulsive or aerodynamic efficiency.

The selection of power rests with the airplane manufacturer, and, ultimately, with the transport operator. The aerodynamic efficiency is in the hands of the airplane designer. But the propulsive efficiency is the concern of the engine manufacturer. It depends upon the propeller efficiency, head resistance, and engine weight in pounds per horsepower. Propeller efficiency can be improved considerably by the use of a geared propeller. That presents no serious mechanical problems but adds somewhat to the weight and cost of the engine. The matter of head resistance opens old wounds, the marks of ancient and honorable battle of the radial versus the in-line engine.

Someone has remarked very aptly that if the young designer were given a basketful of aircraft-engine cylinders with instructions to arrange them in such a pattern that they offer the greatest head resistance and the maximum interference with the pilot's vision, the results could be nothing but a radial engine.

That remark hardly settles the question, but if it be true, as estimated by a prominent aeronautical engineer, that the speed of a 10-to-14-passenger tri-motored airplane can be increased by better than 8 m.p.h. by the use of in-line or V engines, the engine manufacturer has a problem there which demands immediate investigation. We must remember, however, that the radial engine represents the most economical use of material and therefore is somewhat lighter per cubic inch of piston displacement than the corresponding in-line engine. This weight and cost advantage is more marked in the smaller sizes, in which the head resistance of the radial engine also becomes noticeably less important. In general, it seems that, from the point of view of propulsive efficiency alone, the in-line engine would have the advantage in powers above about 200 hp. and that the radial engine will hold its own in lower powers.

Safety Points to Diesel Engines

The use of large transport airplanes gives to the engine manufacturer a new definition of the word "safety." When flying was confined to small airplanes, the engine manufacturer's responsibility, with reference to the safety of the passenger, stopped with the production of a safe, dependable engine. Now, two new factors, radio and the Diesel engine, add to the engine manufacturer's troubles. Safety of the passenger demands constant radio communication with ground stations. This affects the engine manufacturer because radio transmission requires a high-output generator which must be driven by the engine, and because the engine ignition system, unless thoroughly shielded, interferes with radio reception. The generator drive offers no difficulty, but the shielding of the ignition system is something of a task. Radio-shielding of the ignition system requires that every electrical conductor on the engine be completely enclosed with a metallic shield. Even a single spark-plug that is not shielded by some sort of metallic hood makes a perfect little radio station which interferes seriously with radio reception. This problem now seems to be satisfactorily solved and complete radio-shielding equipments are available, at a price, for all types of engine.

The second problem, involving the Diesel engine, is the ancient terror, fire hazard. Fire hazard has always been with us but, when 10, 20, 50 or 100 persons fly in a single airplane, the prospect of a fire in the air or following a crash seems vastly more terrifying than when two or four persons are exposed to the same risk. Anything which even faintly suggests fire hazard in an airplane will keep the public out, and any device which will afford some protection from fire or some assurance that fires are not likely to occur will pay large dividends. Regardless of the origin of fires, it is safe to say that fully 99 per cent of them would never occur were not gasoline or some similar volatile fuel present. It certainly is true that volatility of the fuel is responsible for the instantaneous spread of fires after crashes. The Diesel-engine enthusiasts have some cause for insisting that the Diesel engine will eliminate a large percentage of the crash fires and fires in the air because of the relative non-flammability of the fuel. Perhaps it may be possible to accomplish the same result without the Diesel engine but at any rate the fire-hazard problem demands study. The Diesel engine, if successful, also will make unnecessary the shielding of ignition wires, because it has no ignition wires.

Problem of Eliminating Noise and Vibration

The requirement of comfort in transport planes is by far the most pressing and involved of the four requirements. Air passengers soon discover that flying at the height of 1000 ft. or more over open country is one of the most monotonous methods of spending time. Therefore, having experienced the thrill of flying on the first trip, the passenger on the second trip begins to demand comfort. He is annoyed by uncomfortable seats, dirty windows, excessive noise and vibration and by exhaust odors. He unconsciously compares the airplane with railroad accommodations and insists upon improvements. He wants books and magazines and would like to have radio. He wants less noise and vibration. He insists that the service should be carefully organized, that the airplane be brought to the passenger waiting-room rather than making the passenger go through rain, mud, or snow to get to the airplane. He wants the airplane attractively painted and decorated.

Of all these demands the engine manufacturers' task of silencing the exhaust and eliminating vibration seems to be the most pressing. Two hours in flight in the most uptodate multi-engine airplane will convince the skeptic that opportunity still remains for great improvement in these respects. Some rather ineffective efforts have been made to insulate the cabins from the noise but they serve chiefly to emphasize the fact that the trouble must be attacked at its source.

Fertile Field for Experimentation

Exhaust noise can be silenced, for mufflers are available which will accomplish that nicely. Some of them will appear shortly on transport airplanes. The propeller noise can be reduced considerably by the use of propeller speed-reduction gears. But vibration presents an entirely separate problem, involving again the radial-versus-line-engine controversy. The radial engine has an inherent vibration tendency resulting from our inability to exactly balance the reciprocating and rotating masses and from the variations of stroke and connecting-rod motions due to the method of attaching the link rods to the master-rod end. These unbalanced forces increase with the size of the engine. In properly designed small radial engines they are hardly noticeable. Therefore it seems that the large-engine field is the most fertile field for experimentation, yet all but one of the American in-line engines developed recently with the avowed purpose of reducing airplane vibration are rated at less than 150 hp., the very field in which the least improvement can be expected and the disadvantages of weight and cost of the in-line engine are most pronounced.

We watch with great interest the tests of large in-line engines. We know from earlier experience with 12-cylinder water-cooled engines that perfect balance may be expected, but we still have to determine whether the advantages of smoothness and compactness will be sufficient to offset the disadvantages of slightly increased weight and cost which may be encountered in the large air-cooled or Prestone-cooled in-line and V engines. Probably we shall not be far wrong if we conclude, with the comfort of the passenger chiefly in mind, that the position of the radial engine of more than 200 hp. is precarious. The position of the engine with respect to the passengers also has an important influence on the comfort of the passengers. Unquestionably, vibration transmitted to the passengers through fuselage members is very annoying and will be substantially relieved by the shock-absorbing effect of wing mountings. Direct comparison of recent engines when removed entirely from the fuselage and when mounted in the fuselage will be very interesting.

Demand for Economy Challenges Courage

The air traveler's demand for economy in transportation introduces a discussion of costs that might well run into volumes. However, in the final analysis, it returns to the old vicious circle of production: The public does not fly because the costs are too high; costs are high because the public does not fly. The same condition is largely responsible for the high prices of all aircraft material. The volume is not sufficient to pay for high-production equipment, and without high-production equipment the prices cannot be lowered. Eventually, the problem will be solved by concessions on both sides; an increasing number of people will take to flying despite the high costs; and an increasing number of manufacturers will assume the burden of production equipment and gamble on the success of reduced prices. In high production lies the engine manufacturer's hope of drastic price reductions. Only in that direction can we look for prices at all comparable with the prices of automobile engines. But in the meantime the engine manufacturer can improve his production prospects by keeping one eye at all times on the production possibilities of new designs and adopting such engine types and such methods of construction as afford the best production possibilities.

Many who are not familiar with the problems involved have supposed that the in-line or V engine, because it resembles the automobile engine in general form, should be a much cheaper production job than the radial engine. That is true in production large enough to warrant the use of multiple-spindle drills and boring mills, drum milling-machines, multiple-wheel grinders and other production machinery developed for similar automobile engines. But, in the quantities considered in present aircraft production, the in-line-engine costs would be no lower and perhaps slightly higher than those of the radial engine. However, the difference in costs will not be sufficient to indicate a choice between the two when other characteristics are considered. The responsibility seems to lie on the engine manufacturer to assume the burden of production equipment and to gamble on the success of his design by actually producing engines in quantities as large as the traffic can bear.

Service Requirements Must Be Reduced

One more item of cost in air transport-operation that seems to deserve notice is the cost of engine maintenance. We all have heard the remarkable stories of non-stop engine operation, sealed-engine tests and unbelievable periods of normal operation between overhauls, but a few minutes of heart-to-heart conversation with any well-informed mechanic will convince anyone that the best of aircraft engines requires a considerable amount of nursing. This service is expensive. Whether the in-line or the V engine will improve this condition is still to be demonstrated. To one who is familiar with the service required by present-day aircraft engines, the possibility seems promising; at any rate it is one

(*Concluded on p. 94*)

Heavy-Duty-Motorcoach Brake-Design

By GEORGE A. GREEN[1]

TRANSPORTATION MEETING PAPER[2] *Illustrated with* PHOTOGRAPHS AND DRAWINGS

FOUR-WHEEL brakes are needed on motorcoaches because the demands for higher maximum and schedule speeds and for quicker acceleration impose increased demands on brakes, since rapid acceleration necessitates rapid deceleration. The motorcoach must now provide a performance comparable with that of the average automobile, and this makes the braking problems more difficult. Motorcoach and automobile brake problems are compared, and a logical analysis of the factors involved in the use of four-wheel brakes on motorcoaches is presented.

In analyzing the different systems of braking, both the advantages and disadvantages of the conventional forms of brake-operating mechanism, namely, the mechanical, the compressed air and the hydraulic systems, are listed and commented upon and the merits of the materials used are compared. Metal-to-metal and propeller-shaft brakes also are considered, as well as brake-lining material; but only the principles are referred to, no attempt being made to describe design details.

While real progress has been made respecting brake problems in general, the author states that a large amount of work remains to be done and that, although we seem to have emerged from a condition in which brakes were unsafe to a stage in which reliability can be confidently predicted, designs generally are crude and the weight of the brake systems has been increased tremendously.

In conclusion, future developments of braking systems for heavy-duty motorcoaches are predicted, and their probable general characteristics are stated.

AMONG the many problems concerning the design and operation of motorcoaches, that of brakes must be considered of major importance because of the necessity for providing the element of safety. Great progress has been made in the solution of brake problems, and today the design elements are better understood than at any previous time. Factors which have contributed substantially toward progress are the development of reliable air-brake controls and improved brake-drum and brake-lining materials.

In considering why four-wheel brakes are necessary, it may be said that the demand for higher maximum and schedule speeds and quicker acceleration impose still further demands on brakes, because more rapid acceleration necessitates more rapid deceleration. The motorcoach must now provide a performance comparable with that of the average automobile and, since the coach must hold its place on the road with the passenger-car, the demand is not unreasonable; but, among others, the braking problems are greatly increased. Laws which specify minimum stopping distances have been enacted in a number of States; for example, those in some States require a 35-ft. stopping distance from a speed of 20 m.p.h. Such a stop is equivalent to a deceleration of 12.3 ft. per sec. per sec. but, since 2.4 sec. are required to make the stop, it seems impractical, solely from the viewpoint of safety, to consider restrictions of a less arduous nature.

Considering the foregoing typical case and assuming a vehicle weight of 20,000 lb., including a full complement of passengers and baggage, such a stop would require an energy absorption of 267,500 ft-lb., which is equivalent to 204 hp. A matter of considerable importance is that the energy absorbed as a result of brake applications must be replaced in the form of fuel, and this emphasizes the necessity for educating drivers to appreciate that a very real gasoline saving can be effected by careful operation. The point is that better fuel economy is obtainable as a result of consistent moderate speeds than if bursts of speed are indulged in, because high speed almost invariably necessitates needless brake applications, and consistent speeds rather than bursts of speed tend toward higher average and safer schedule speeds. It also must be remembered that decelerative energy expressed in terms of fuel is necessary even in larger quantity to provide the ensuing acceleration. Actually, more energy is needed to accelerate than to decelerate because of the losses in efficiency throughout the powerplant, transmission and axles.

Motorcoach and Automobile Brake Problems Compared

In comparing the brake problems of motorcoaches with those of automobiles, it is interesting to note that, considering some of the more prominent makes of automobile and taking into account full loads in both cases, there is from 15.6 to 17.6 lb. of automobile weight per square inch of service-braking area, but as high as 75 lb. in the case of the motorcoach. The trend in coach design generally is toward greater luxury, which means weight added to both the body and the chassis. Two types of heavy-duty motorcoach are shown in Figs. 1 and 2. To simplify maintenance costs, virtually all chassis parts are being made larger and heavier and, in many instances, this substantially simplifies their design. It also has been necessary to increase the size of the tire equipment to take care of the additional weight, and this has necessitated the development of rims of smaller diameter, which in turn reduces the amount of available brake-drum space.

Balloon tires are becoming increasingly popular and it seems reasonable to suppose that soon all heavy-duty passenger-carrying vehicles will be equipped with them. A typical rear-wheel assembly is shown in Fig. 3. For the automobile, the balloon tire has entirely replaced the high-pressure type and experience indicates that the

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[2] Prepared for presentation at the World Engineering Congress, Tokio, Japan, Oct. 29 to Nov. 22, 1929.

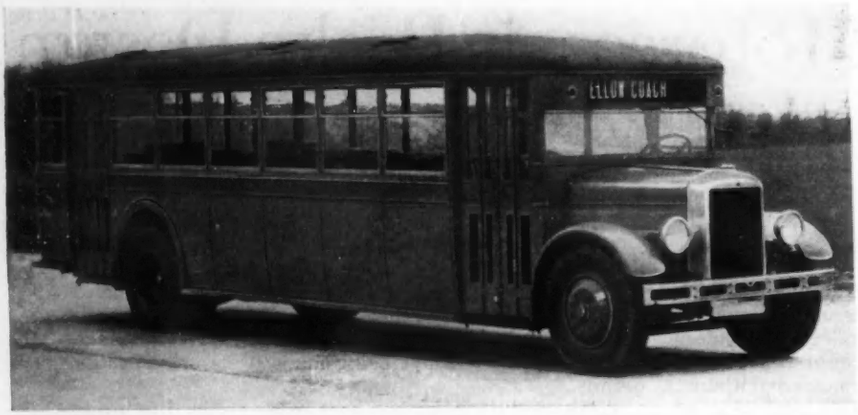


FIG. 1—HEAVY-DUTY CITY-SERVICE 40-PASSENGER MOTORCOACH

trend of coach design generally is to follow that of the automobile; but the standardization of balloon tires influences brake design because there is a practical limit beyond which the outside diameter of the tire cannot be increased. Having determined this dimension, it will be found that the inside diameter must be reduced as compared with a high-pressure tire capable of carrying an equal load, and this further restricts the space available for the brake mechanism.

It obviously is desirable to make the brake-drum diameter as large as possible, but an excessively large drum requires large tires, large wheel-pockets, a high frame, a high center of gravity and high steps, and operators are opposed to all of these features. The adoption of a standard diameter for all brake-drums evidently is desirable from both production and service viewpoints; but, in determining this dimension, it is necessary to take into account the fact that either balloon or high-pressure tires may be used, the maximum size of such tires to take care of the loading, the rim equipment required, and the space between the rim and the brake-drum, that is, the air-gap, which has a large influence on the life of the tire.

Practical experience has demonstrated that there should be a clearance of at least 1 in. between the brake-drum and the rim and, if this clearance is provided, that 40 x 9.75-in. balloon-tires, which meet the average conditions for either heavy-duty city-service or parlor-car operation, should be used. Assuming a conventional design, the clearance between the rim and the brake-drum, the size of the rim and the size of the brake-drum are automatically determined; in short, the drum diameter will be 17¼ in., the clearance between the rim and the brake-drum 1¼ in., and the rim diameter 22 in. On the assumption that the largest permissible brake-drum diameter is 17¼ in., it obviously is necessary to provide brakes on all four wheels to assure a satisfactory braking system.

When considering the elements which determine the heavy-duty-motorcoach maximum permissible brake-drum diameters, it is interesting to reflect upon the fact that, with the larger automobiles, brake-drum diameters range from 16 to 16½ in. and that brake problems on the average automobile are by no means solved; further, that the laden weight of the larger automobile as compared with the heavy-duty motorcoach is in a ratio of approximately one to five. No matter how one may regard the situation, it is evident that the limitations surrounding heavy-duty - motorcoach brake - design necessitate taking every possible advantage of available braking areas and, with the multi-wheel systems of six or eight wheels, the possibility of obtaining more satisfactory and more economical operation as a result of applying brakes on all wheels is of particular interest.

FIG. 2—HEAVY-DUTY INTERURBAN 29-PASSENGER PARLOR-CAR MOTORCOACH

Four-wheel brakes are today almost universally employed on the heavy-duty motorcoach and it may perhaps be thought unnecessary to argue their merits; but it is believed that a logical analysis of the factors involved may be of interest and, while real progress has been made respecting the solution of brake problems generally, a very considerable amount of work remains to be done. In fact, we seem simply to have emerged from a condition in which our brakes were unsafe to a stage in which reliability can be confidently predicted; but designs generally are crude and weights have been tremendously increased. These comments apply especially to the brake-drums and shoes.

The Four-Wheel Mechanical Brake

Originally, all heavy-duty motorcoach and four-wheel-brake systems were of the so-called mechanical type in which the force of application was transmitted through flexible or non-flexible metallic connections and the designs in general followed current automobile practices. High pedal-pressure was the rule rather than the exception in the earlier days, and various power-intensifying devices such as the mechanical servo, vacuum and air booster were developed. In the light of present experience it seems impossible to furnish a sufficiently low pedal-pressure without power application and, while experiments are being made with the light and the medium-duty motorcoaches on which the required servo or self-actuation is provided in the shoe mechanism, the

progress of this work to date is not sufficiently advanced to forecast accurately the future possibilities, although the indications are that such an arrangement is impractical insofar as it concerns heavy-duty equipment.

A point worthy of note is that in all cases power application of brakes has necessitated a very considerable strengthening of the various levers and cross-shafts to assure freedom from twisting or bending and, when cross-shafts are of considerable length, a diameter of 3 in. or even 4 in. may be necessary. But even if light-gage tubes are employed, the cost and weight of the tubes, levers, brackets and bearings are factors which cannot be ignored.

An advantage of the mechanical braking system is that the connections are readily traceable to their sources without the necessity of undergoing any imaginative processes; but considerable experience is needed to determine just where the adjustments should be made, especially if equalizers are employed. The disadvantages of mechanically actuated brakes are as follows:

(1) The most serious disadvantage lies in the geometrical problems which exist in connection with both front and rear brakes, these comments applying particularly to six and eight-wheel motorcoaches. It is necessary to take into account both the up-and-down and the fore-and-aft wheel-movement; in some cases, rear axles travel in the vertical plane 14 in. from full rebound to full bump, and the maximum front-wheel angularity may be as much as 45 deg. To assure satisfactory brake operation under such conditions necessitates the solution of a number of problems and, unfortunately, their solution usually results in compromises of one sort or another.

(2) While adjustments can be made which give satisfactory results, considerable skill is needed to make them.

(3) Numerous rods, cross-shafts, bearings, yokes, clevises and other parts requiring adjustment demand constant care and attention.

(4) If the cross-shafts and joints are not properly lubricated they tend to seize and wear, which causes undue noise, increased frictional resistance and resultant higher pedal-pressures.

(5) The large number of attached parts increases the possibility of failure; for example, the rods may break, the clevis pins come out, or the adjustments become loose.

(6) The cross-shafts, being of necessity large in diameter, require large levers, bearings, brackets and attaching parts, and this increases both cost and weight.

(7) To provide reasonable pedal pressure, some form of power application is essential.

All four-wheel brakes originally were of the so-called mechanical type, but this arrangement has steadily lost in popularity; today, only a very limited number of the large-scale manufacturers advocate their employment, which seems to indicate that the disadvantages outweigh the advantages.

FIG. 3—A TYPICAL REAR-WHEEL ASSEMBLY

This Assembly Weighs 388 Lb., the Weight of the Tire Alone Being 135 Lb., and Includes the 40 x 9.75-In. Balloon-Tire, the 22-In.-Base Rim, the 17¼-In. Brake-Drum, the Oil Slinger, the Disc Wheel and the Hub

The Four-Wheel Air-Brake

In the early days of the motorcoach few, if any, vehicles possessed what could be described as a good braking system; but, since there is ample precedent for good brakes in connection with the operation of steam and electric traction lines, it seems natural that consideration should have been given to the incorpora-

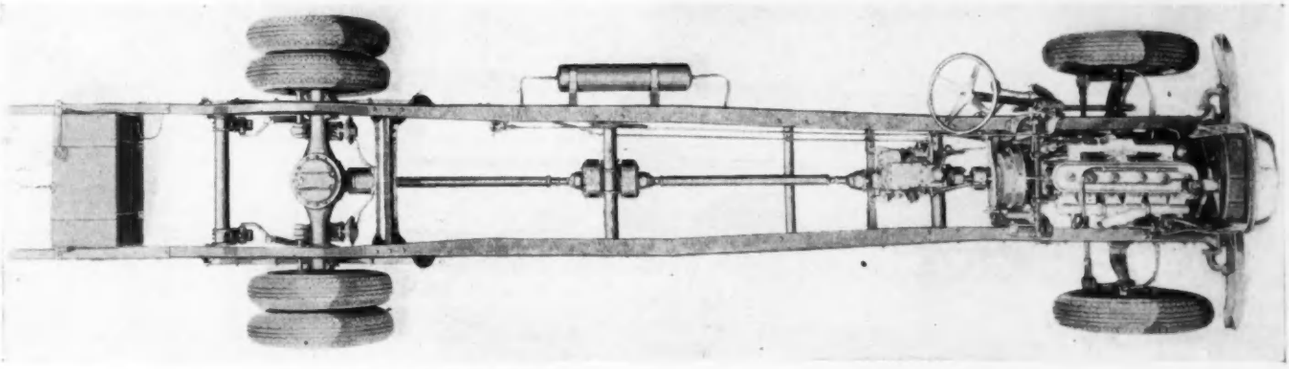


FIG. 4—HEAVY-DUTY-MOTORCOACH CHASSIS EQUIPPED WITH FOUR-WHEEL AIR-BRAKES

tion of the air-brake system in the motorcoach. A heavy-duty-motorcoach chassis equipped with four-wheel air-brakes is shown in Fig. 4.

The Westinghouse Air Brake Co. was the pioneer of the motorcoach air-brake system. While this company's first efforts to develop and introduce motor-vehicle air-brakes were not particularly successful, air-brake operation is now almost universally standardized for the heavy-duty motorcoach. Many years of precedent lie behind the development of the compressor, control and relief mechanism and, while in the earlier stages mistakes were made in the design of this apparatus for motorcoach application, particularly with respect to the fragility of the parts, most if not all of the original defects have now been eliminated. Fig. 5 illustrates a typical air-compressor installation.

Many serious brake problems developed on the Pacific Coast with respect to fast long-distance services. The older forms of brakes were incapable of holding vehicles safely on long grades, and it can truthfully be said that the Pacific Coast pioneered the development of what is believed to be the most successful brake-operating medium that the heavy-duty-motorcoach industry has as yet seen; namely, compressed air. In the light of present knowledge the advantages of the air brake appear overwhelmingly to outweigh the disadvantages, as is apparent from the following listings:

Advantages of Air Brakes

(1) No geometrical problems exist in connection with either front or rear wheels.

(2) Minimum physical effort is required on the part of the driver.

(3) The effort required for each brake application is in direct proportion to pressure and pedal movement; that is, there is a definite sense of "feel."

(4) The equalization at all four wheels is automatically exact.

(5) The system automatically provides the compressed air required for the operation of doors, fare-registering devices and warning signals, which are now the rule rather than the exception.

(6) The largest potential motorcoach users, namely, the operators of steam railroads and of electric-traction lines, are accustomed to their use and understand and appreciate many of the problems involved.

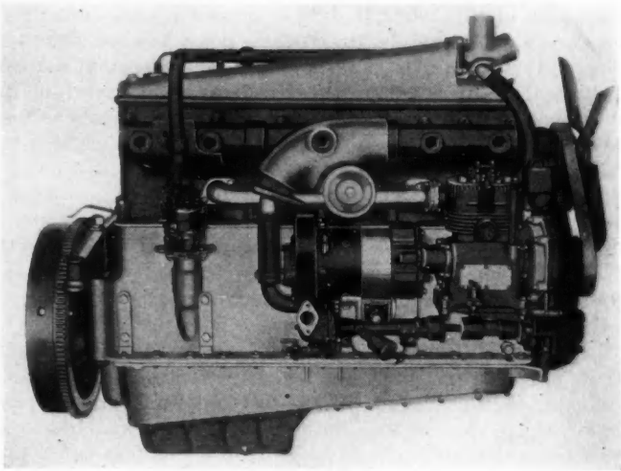


FIG. 5—A TYPICAL AIR-COMPRESSOR FOR MOTORCOACH INSTALLATION

Disadvantages of Air Brakes

(1) The tracing of troubles to their source and the making of adjustments requires experience.

(2) The failure of any of the connections means a loss of the air pressure, under which circumstances the service brakes must necessarily become inoperative.

(3) A good four-wheel air-brake is much more expensive than any other form of brake mechanism, but it is felt that safety must be secured at any cost.

(4) The mechanism is complicated but in its present stage of development has proved to be relatively free from trouble. In spite of the complications, its reliability seems to compare favorably with that of any of the auxiliaries such as the starting, lighting and ignition systems, and, as the manufacturing company maintains service throughout almost all the civilized world, it usually is possible to obtain skilled advice and attention.

The Four-Wheel Hydraulic Brake

Hydraulic brakes have been employed thus far only on the light and the medium-duty motorcoaches; but a number of passenger-car and light-truck manufac-

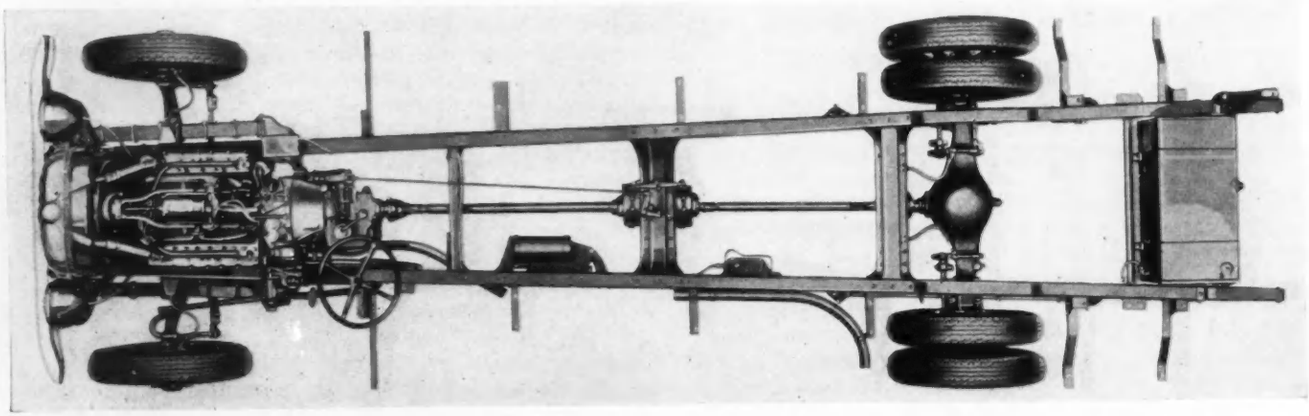


FIG. 6—MEDIUM-DUTY-MOTORCOACH CHASSIS EQUIPPED WITH FOUR-WHEEL HYDRAULIC BRAKES

turers have standardized on this system and today, taken as a whole, very successful results are obtained. Furthermore, on account of the increased use of this mechanism, maintenance problems are better understood and service is available throughout the entire Country. A medium heavy-duty-motorcoach chassis equipped with hydraulic four-wheel brakes is shown in Fig. 6.

With the hydraulic braking system the rods and their connections are replaced with liquid columns operated by a simple form of ram, this arrangement being ideal because the liquid is incompressible and no frictional losses occur. At first sight this system may seem to provide no mechanical advantage, but this is not true because the elimination of the cross-shafts and their bearings materially decreases frictional resistance. Another very important point is

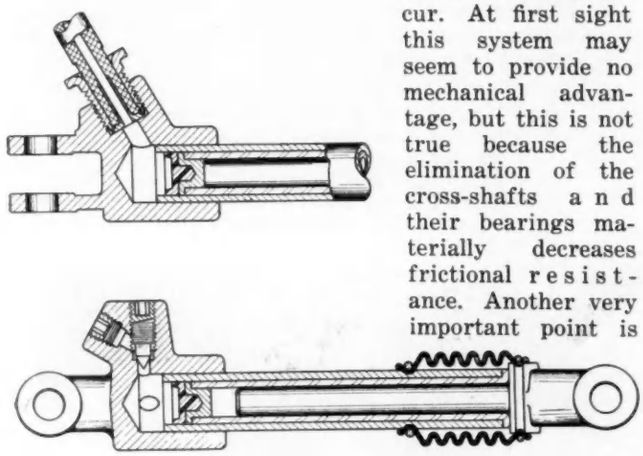


FIG. 7—SECTIONAL VIEW OF AN EXTERNAL BRAKE-OPERATING CYLINDER

that the efficiency of an hydraulic braking system remains constant, which is not the case with the mechanical brake because the connections, shafts and bearings may and often do become tight owing to lack of lubrication, a condition which, unfortunately, often obtains. Obviously, this results in a loss of efficiency which must be compensated for by means of increased pedal-pressure.

Advantages of Hydraulic Brakes

(1) No geometrical problems are involved in connection with either front or rear wheels.

(2) Lag and lost motion are at the minimum and, in these respects, the operation is superior to that of both mechanical and air brakes.

(3) The equalization at all four wheels is automatically exact.

(4) From a manufacturing viewpoint, the mechanism is less costly than either air or mechanical systems.

(5) The weight of the mechanism is less than that of either air or mechanical systems.

Disadvantages of Hydraulic Brakes

(1) A special operating fluid is required and, if the fluid composition is not correct, the entire system is likely to fail; but this fluid is not expensive and can be obtained from all authorized brake-service stations.

(2) The making of adjustments requires experience, particularly with regard to the operation of eliminating air from the lines, a process known as "bleeding"; but it is believed that hydraulic-brake adjustments in general require considerably less skill than does an air-operated system.

(3) To provide moderate pedal pressure, some form of power application is essential; but in this respect the conditions do not differ from those of the mechanical brake.

(4) Operators of steam and electric traction lines are in general unacquainted with this form of mechanism and this, from a merchandising angle, cannot be ignored.

(5) Failure of any of the connections causes a loss of fluid, which makes the service brakes inoperative. This may seem to be a serious indictment, but the same condition obtains with the air brake except that loss of air and not of fluid is involved, and a more or less similar state of affairs obtains with the mechanical brake in the case of rod breakage or clevis-pin failure.

A factor that now seems to make possible the successful use of hydraulic brakes for heavy-duty motorcoaches is the comparatively recent development of external brake-operating cylinders which, because of their exposed position, are not subject to damage by heat and can be attached and detached readily. Their construction is simple, as shown in Fig. 7, and they are light and inexpensive. A front-wheel hydraulic-brake with an external brake-operating cylinder in position is shown in Fig. 8. A combined booster and hydraulic master-cylinder unit which simplifies the hook-up considerably is now available also, the Westinghouse type of this unit being shown at the top in Fig. 9 and that of the Bragg-Kliesrath Corp. at the bottom.

Compared with either the air or the mechanical brake, the hydraulic brake seems to have the advantage as regards simplicity. No serious problems of installation are encountered, and chassis equipped with hydraulic brakes have a neat and attractive appearance. An argument in favor of air-operated rather than mechanical or hydraulic brakes is that compressed air is desirable for other purposes such as the opening and closing of doors and the operation of fare-registering devices,

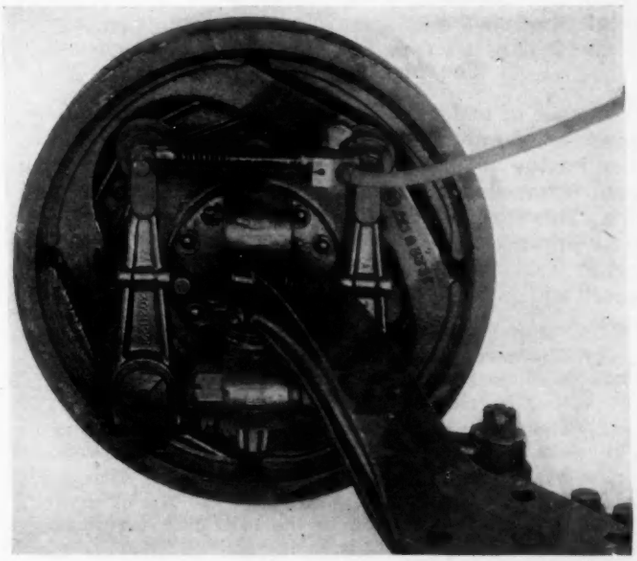


FIG. 8—FRONT-WHEEL HYDRAULIC-BRAKE INSTALLATION

The External Brake-Operating Cylinder Is Shown in Position. Note That the Design Provides for 100-Per Cent Contact of the Shoes with the Drum as Nearly as That Is Possible and for Uniform Pressure around the Entire Interior Circumference of the Drum

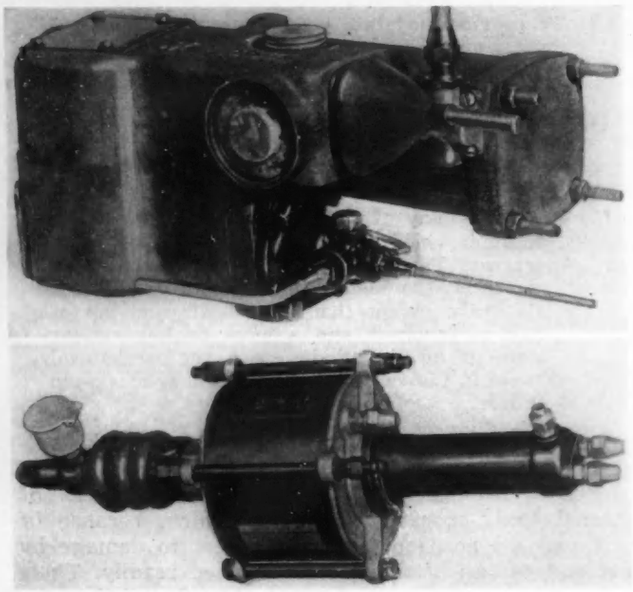


FIG. 9—TWO TYPES OF VACUUM BOOSTER AND MASTER CYLINDER CONSTRUCTED AS AN INTEGRAL UNIT

The Type Shown at the Top Is That of the Westinghouse Air Brake Co.; That Below Being Manufactured by the Bragg-Kliesrath Corp.

horns and other types of warning signal, although either mechanical or hydraulic brakes can be operated successfully with air pressure supplied from a small compressor and storage tank, instead of using a vacuum booster.

It is believed that the advantages far outweigh the disadvantages and that, in the future, the use of the power-operated hydraulic-braking system will be extended to heavy-duty motorcoaches. The importance of minimum lag is a factor of considerable moment where multi-wheel braking is considered, as in braking six or eight-wheel vehicles.

The Metal-to-Metal Brake

The use of metal-to-metal brakes was originally sponsored by operators of street-cars and steam railroads, they having in mind the satisfactory service obtained from external-shoe brakes regularly used in their service. This condition is ideal but, unfortunately, it seems to be impracticable for the motorcoach. The success of metal brakes when used on the exterior surface of a wheel which revolves on a steel roadbed is dependent upon

(1) The hardening of the rim or area of brake-shoe application resulting from the rolling contact of the rim with the rail.

(2) The tendency for the exterior surface of the rim to become smooth as a result of its rolling contact with the rail.

(3) The rapid transfer of heat from the rim or area of shoe application to the rail, as a result of its rolling contact with the rail.

(4) The rapid atmospheric heat-transfer from the wheel and shoe as a result of the exposed position of these parts.

(5) The large amount of metal in the wheel and shoe, which provides an almost unlimited heat-storage capacity.

(6) The lack of dimensional and weight restrictions, which permit the provision of a very large body of metal to absorb heat and resist wear.

The lessons learned from the use of metal-to-metal brakes on trains and street-cars are of interest, but it is not believed that any entirely satisfactory method has been found as yet for their application in motorcoach operation, the primary reasons being that

(1) None of the advantages provided by a steel road-surface are available.

(2) The space in which the brake mechanism must be located is, generally, small and poorly ventilated.

(3) The advent of the balloon tire and the small-diameter rim has increased the heat-dissipating problems greatly.

(4) Trains and street-cars are provided with external brakes, which system is not adaptable to the motorcoach; first, because the exposed position of such brakes would result in rapid wear on account of the accumulation of dirt, sand or gravel; second, because it would be very difficult to provide a brake of suitable diameter since insufficient space is available for the operating mechanism and, if the brake mechanism were moved inward, it would be necessary to reduce the frame width to less than the limits that are considered practical.

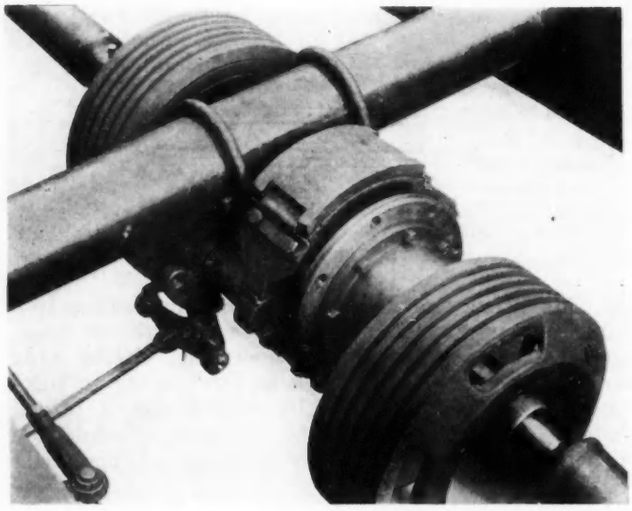


FIG. 10—HEAVY-DUTY TWIN INTERNAL-TYPE PROPELLER-SHAFT-BRAKE ASSEMBLY

The Drums Are Mounted on Either Side of the Propeller-Shaft Center-Bearing. One Drum Is Shown Slid Back To Illustrate the Brake-Shoe Construction, the Molded Liner and Its Attachment

Disadvantages of Metal-to-Metal Brakes

The advantage of metal-to-metal brakes is that the shoes possess excellent heat-absorption properties, particularly the so-called throw-away type; but six disadvantages, which apply particularly to the throw-away type of brake-shoe, are that

(1) The friction coefficient of metal varies with the speed; consequently, when the brakes are applied, the wheels tend to lock as the vehicle comes to rest.

(2) The difficulty of assuring a smooth stop represents a definite accident hazard.

(3) Their harsh operation has a very bad effect on the equipment in general.

(4) Considerably greater care is required in connection with their operation, as compared with the conventional woven or molded brake-linings.

(5) The life of both shoes and drums is relatively short and is shortened further by careless operation; cutting and scoring of both drums and shoes is the rule rather than the exception.

(6) The renewal cost of both drums and shoes is excessive.

Marked differences of opinion have existed, and still exist, regarding the advantages of metal-to-metal brakes as compared with the various types of brake-lining. Until comparatively recently the metal-to-metal brake, air operated, was given very general preference; but this attitude seems to be changing, chiefly as a result of the availability of improved brake-drum and brake-lining materials, specifically close-grained cast-iron drums and molded or block-type linings. It is felt that the advantages of the metal-to-metal brake existed more in the minds of operators than in fact, and that the reasons underlying the success of this form of brake, in its application to the steam and electric railroads, were not properly understood; further, that the difficulty of obtaining the same conditions with the heavy-duty motorcoach were not appreciated. From the arguments set forth, the disadvantages of metal-to-metal brakes seem overwhelmingly to outweigh their advantages.

The Propeller-Shaft Brake

Whether propeller-shaft brakes are to be used or not continues to be a much-discussed subject, and there have been many reversals of opinion. Originally, nearly all heavy-duty motorcoaches were provided with propeller-shaft brakes, largely because of the unreliability of the old-style conventional rear-wheel mechanical brakes. A heavy-duty twin-drum internal-type propeller-shaft-brake assembly is shown in Fig. 10, and one for medium duty in Fig. 11. A heavy-duty front brake-shoe assembly, including molded liners, a high-lift cam, and the like, is shown in Fig. 12.

The joint use of propeller-shaft and air or hydraulically operated brakes has specific advantages; namely,

(1) The elimination of cross-shafts, their bearings, supporting brackets and attaching parts, since it is difficult to provide cross-shafts of proper size to resist bending or twisting without adding a large amount of weight. If cross-shaft bearings are not kept properly lubricated they tend to seize, with disastrous results in case the brakes become inoperative.

(2) Rods, cables, clevises and clevis-pins, which parts tend to wear rapidly and cause noise, particularly if they are not properly lubricated, are eliminated; in addition, the danger always exists of rod and cable breakage, and that clevis-pins may come out.

(3) The elimination of turnbuckles or similar adjusting means, because lessening the number of adjustment points reduces wear and possible maladjustment.

(4) The simplification of rear-brake adjustments where there are no mechanical connections is of considerable importance. Where the rear brakes are operated by rods or cables, long leverages are necessary; the ratio of movement between the hand lever and the cam is marked, and this results in the necessity for adjusting the rear-brake shoes frequently to make certain that the emergency brake is operative, which, in turn, requires the maintenance of close clearances between the brake-shoes and brake-drums. If the adjustments are not very carefully made, dragging will result and cause rapid drum and lining wear, the unnecessary use of fuel, and possible damage to tires from excessive heat. All of these troubles are minimized when there is no mechanical connection to the rear shoes, and the adjustment of the shoes is so simple that it can be taken care of by a comparatively inexperienced man, particularly bearing in mind that the brakes, being operated by air or liquid, are equalized automatically and relatively wide clearances be-

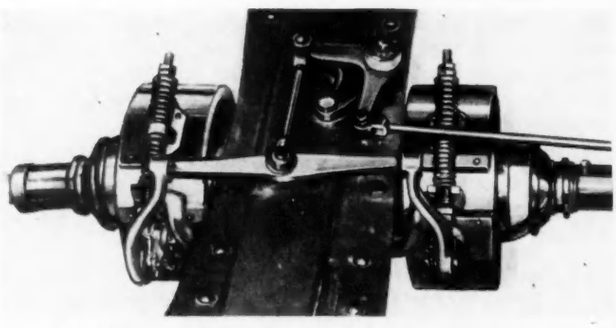


FIG. 11—MEDIUM-DUTY TWIN-DRUM EXTERNAL TYPE OF PROPELLER-SHAFT-BRAKE ASSEMBLY WITH THE DRUMS MOUNTED ON EITHER SIDE OF THE PROPELLER-SHAFT CENTER BEARING

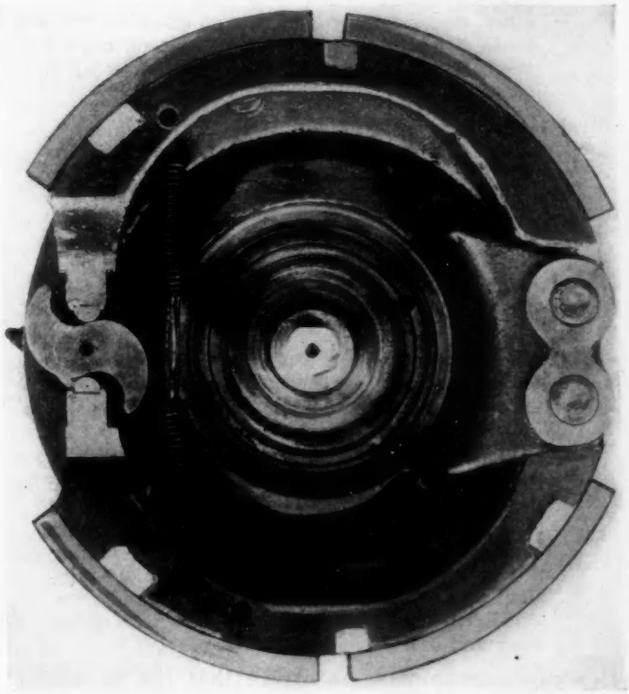


FIG. 12—HEAVY-DUTY FRONT BRAKE-SHOE ASSEMBLY INCLUDING MOLDED LINERS, A HIGH-LIFT CAM, AND OTHER PARTS

tween the drums and shoes can, with safety, be maintained.

(5) Where there are no mechanical connections to the rear shoes, no geometrical problems exist; that is, the clearance between the brake-shoes and brake-drums will remain constant regardless of the amount of weight carried. The difficulty of obtaining exactly the right lever lengths and locations to assure that the brakes do not come on or off under increasing or diminishing loads is a very real one; in nearly every design, this is a matter of compromise.

(6) From a legal viewpoint the standardizing on a propeller-shaft brake is a decided advantage since, under these conditions, no question arises as to the legality of the brake system, regardless of present or future legislation, local or otherwise.

Disadvantages of Propeller-Shaft Brakes

(1) The drum or drums on the propeller-shaft must be balanced very carefully; otherwise, shaft vibration will occur.

(2) It is difficult to provide adequate heat dissipation and a drum of adequate size to assure reasonable brake-lining life; but, assuming careful design, it is believed that these problems can be solved.

(3) Unless the design is carefully developed, the drum or drums may project above the top of the frame side-bars and this, from the viewpoint of body design, is troublesome.

(4) Additional stresses are imposed on the propeller-shafts and rear-axle-drive mechanism and proper provision must be made to take care of this condition. The use of a propeller-shaft brake should be considered only after assuming the existence of reliable four-wheel-brake equipment; otherwise, the propeller-shafts and rear-axle-drive mechanism will be subject to undue wear and tear.

It seems unreasonable to expect that the propeller-shaft brake should be used as a service member, and it is anticipated that its employment will be confined to emergencies, parking, and holding vehicles on grades while passengers are entering or alighting.

The trend of domestic legislation is to demand two entirely separate braking systems; but to write this into a law is not at all simple and, while many attempts have been made to do so, the language is still open to misinterpretation. Consequently, the only certain method is to provide two entirely separate systems not interconnected in any way whatsoever. It seems reasonable to suppose that, ultimately, laws will be enacted which will make the installation of the propeller-shaft brake compulsory.

Fabric and Molded Brake-Linings

Molded and fabric brake-linings are the two general types used in addition to the metal linings previously discussed. Both types include a wide range of product and, incidentally, a multitude of defects. Long precedent and low first-cost seem to be the only reasons for the present employment of woven linings, which have the following disadvantages:

(1) No two manufacturers make the same kind of lining, and all have different views as to what constitutes a good lining.

(2) It is difficult, if not impossible, to obtain a woven lining of sufficient thickness to provide a reasonable length of life. As a rule, thick brake-linings are compressible and this interferes considerably with satisfactory brake operation.

(3) Most woven linings are impregnated with various substances which are partly or completely inflammable and give off unpleasant odors when hot.

(4) The common use of wire as a binder frequently causes scored brake-drums and this, because of the necessity for remachining them, decreases their length of life.

Advantages of Molded Brake-Linings

(1) The thickness of molded brake-linings may be considerably in excess of that of the woven type, since they are not compressible in anything like the same degree.

(2) Because of the greater thickness, the method of fastening them is relatively easy; for example, bolts can be used instead of rivets and this simplifies their renewal. If the bolted-on method is not considered satisfactory, integral keys or other registering devices of various forms can readily be incorporated in the liners. The elimination of rivets greatly minimizes the possibility that the brake-drums will become scored.

(3) It is not necessary to use wire or any form of metallic fiber as a binder, which also minimizes the possibility of drum scoring.

Disadvantages of Molded Brake-Linings

(1) In general, the friction coefficient is less for molded than for the woven type of lining, but this can be compensated by the provision of additional pressure.

(2) The first cost is considerably in excess of that of the woven type, but it seems reasonable to expect that generally better brake preformance, including longer life and the absence of drum scoring and brake grabbing, will amply compensate for this.

Molded brake-linings seem to overcome almost all of the disadvantages of the fabric brake-linings, and it seems reasonable to suppose that the thick bolted-on type soon will be in general use.

Summary

An attempt has been made in this paper to compare in an impartial way certain of the fundamentals of the more important conventional types of heavy-duty-motorcoach braking-systems; but such an analysis is incomplete without an attempt to forecast future developments even though such a forecast necessarily must be merely a matter of opinion. However, I believe that the future heavy-duty-motorcoach braking-systems will possess the following general characteristics:

(1) Hydraulic operation, with either pressure or vacuum-type boosters
(2) A booster and master cylinder as an integral unit
(3) Hydraulic brake-operating cylinders of the external type
(4) Brake-shoe design that provides for 100-per cent drum-contact as nearly as possible, with uniform pressure around the entire interior circumference of the brake-drum
(5) The molded type of brake-linings
(6) Propeller-shaft brakes
(7) Brakes on all wheels

Fuel Specifications as Viewed by the Fleet Operator

By Adrian Hughes, Jr.[1]

Pennsylvania Section Paper

INCREASING operation of large fleets of motor-trucks and motorcoaches will, according to the author, have an increasingly greater effect on fuel specification, the kind of fuel produced, and on engine design. Conditions met by the operator in the purchase of fuel and the practical limitations of fuel specifications are surveyed. The requirements that a fuel must meet to be satisfactory to the operator are set forth and the four kinds of fuel generally available are listed.

Although the results that are required from a fuel are well known, a clear understanding of the relations between the operating results and the make-up of the fuel has been reached only recently by the specialists. Even if operators were conversant with the technical facts, they would still be faced with the difficulty of specifying tests that would assure the necessary qualities and that could be performed conveniently in practice.

Experience of the motorcoach operating company with which the author is connected, with fuels used since it began operations in 1915 is narrated, and the specification adopted in 1928 for a benzol-blended gasoline is given. A two-year program of operating tests on two groups or motorcoaches is now under way to ascertain the most suitable type of fuel. Considerable importance is attached to the use in these tests of a closed water-cooling system which heats up quickly and maintains considerably higher operating temperatures than those of the conventional cooling system.

Research that has been carried on cooperatively by the petroleum and the automotive industries is credited with having resulted in undoubted improvement in fuels and in the efficiency and economy of internal-combustion engines. Similar close cooperation of the engine and petroleum engineers with the fleet operators is urged, to the end that the operators can secure the most satisfactory operation with the lowest cost for fuel. The petroleum industry is asked to formulate standard specifications for the several grades of fuel so that they shall be uniform, and the engine manufacturers are asked to recommend the specific kind of fuel best suited to each engine model.

DISCUSSION of fuel specifications from the viewpoint of the purchaser should be helpful in the efforts that have long been under way to improve economies in the propulsion of vehicles with crude-oil products. Today, few operators know what kind of fuel they need, and none of them knows how to specify the fuel he wants to purchase. Most of them, however, are dissatisfied with the situation, and many of the larger operators are making efforts to obtain definite information about fuel and fuel performance with a view to finding a remedy for it. The subject is highly technical, and the average operator experiences great difficulty in attacking it. Frank discussion by representatives of the oil refiners, the engine manufacturers and the operators so that each shall understand the requirements and the problems of the others, followed by a sincere effort to cooperate, is certain to result in an improvement in the situation that will be satisfactory to all.

In the operation of motor-trucks and motorcoaches, large-fleet operation is becoming the rule rather than the exception. This tendency is increasing and undoubtedly will reach large proportions and have an increasingly greater effect on fuel specification, on the kind of fuel produced by the refiners, and on engine design. The larger a fleet operation becomes, the better it will be equipped with capable engineers who will carry on reliable experiments and research leading to more satisfactory service and greater economies. Also, it will mean that both fuel and equipment will be purchased in larger quantities.

[1] M.S.A.E.—Superintendent of bus transportation, United Railways & Electric Co. of Baltimore.

The object in this paper is not to attempt to give a technical discussion of fuel but rather to present conditions which are met by the operator in the purchase of fuel and the practical limitations of specifications in this connection. The technical features of fuel production and engine design can best be handled by the engineers who have specialized in this study. Valuable information already has been contributed by them, and their research and experimenting is continuing. We are more concerned here with the practical application of their findings.

The Operator's Fuel Requirements

For an operator to state definitely his requirements for a fuel is comparatively easy. They are as follows:

(1) Economy; that is, high mileage per gallon of fuel

(2) Power, both for accelerating and for heavy pulling. This includes smooth engine-performance

(3) Freedom from knocking

(4) Low engine-maintenance cost and long life of engine. This includes the minimum dilution of the lubricant and the minimum carbon formation, which interferes with economy, power and long life, and causes interference with service for frequent cleaning.

However, the operator finds it very difficult to prepare a specification that can be submitted generally for bids in an open competitive market and that will assure his obtaining the required qualities in the fuel. Although the results that are required from the fuel are well known, only recently has any clear understanding been reached as to the definite relations between these operating results and the make-up of the fuel even by

the specialists. Even among them some of the points have not been entirely cleared up. Operators in general are not conversant with these technical facts and if they were they would still be facing the difficulty of specifying tests that would ensure the required qualities and that could be performed conveniently in practice.

One of the complications in determining the qualities a fuel should have is that it must function under many extreme conditions. No other device is required to operate under such extremities of load or temperature as does the automotive engine. As such conditions cannot be avoided, it appears at present that the fuel must have the necessary qualities to meet them. Another difficulty met in practice is that, as nearly every fleet consists of groups of vehicles purchased at different times and from various manufacturers, the engine characteristics of each group may vary widely. Therefore, to obtain satisfactory operation and the maximum economy, several grades of fuel might be required.

Baltimore Motorcoach Fleet Experiences

The fleet of the Baltimore Coach Co., which is the motorcoach-operating subsidiary of the United Railways & Electric Co. of Baltimore, consists of 122 vehicles, all mechanically driven. The fleet includes engines of eight distinct types or models. The operation is entirely within the city with the exception of the chartered business, and covers about 2,500,000 motorcoach miles per year. From 1915, when operation was started in Baltimore, up to 1924, gasolines were purchased in the open market without the use of a specification. From 1915 to 1919, the gasolines showed a decrease in performance, as ascertained by the observations and mileage records of the operating superintendent. As is well known, the gasolines produced in the years 1917 to 1919 inclusive were rather unsatisfactory. Trouble experienced during that period is now known to have been due largely to heavy ends of the fuel not being burned dry. As a result, dilution was excessive and numerous difficulties followed.

At about that time benzol blends appeared in Baltimore. Various gasolines had been tried but a 50-per cent benzol blend seemed to give the best performance, and engine knock was eliminated or at least reduced. The fleet then consisted of only 26 buses, all of one model. In 1924, with the advice of a firm of local chemists, a specification calling for a 42-per cent benzol-blend fuel having an initial boiling-point of 132 deg. fahr. was prepared, as it had been concluded from experiments that such a fuel would not produce any appreciable engine knock. This fuel proved rather satisfactory, the price differential was not great in comparison with straight gasoline, and an appreciable saving in engine maintenance and carbon cleaning seemed to be apparent. Experiments in 1927 with the straight gasoline then available, which was very superior to that produced eight or nine years before, showed that the cost per vehicle mile for fuel was probably slightly higher with the benzol blend but it was considered that the saving elsewhere justified its use.

Present Motorcoach Fuel Specification

In the fall of 1928 a revised specification, in which the benzol was reduced to 35 per cent, was adopted. Blended fuels were being used more commonly by oil companies and this reduction in benzol content was made for the sole purpose of trying to get into a more competitive market. That this was rather difficult to do at that time probably is generally known and we learned.

The revised specification reads in part as follows:

> This 35-per cent special motor fuel to be furnished by the oil company shall not only contain 35 per cent benzol, but the gasoline content shall be a special California grade which itself contains some of the antiknock properties of benzol.

The oil company shall use this California gasoline in the 35-per cent blend, and, by the addition of some natural gasoline, the 35-per cent blend shall meet the following specifications:

(1) It shall contain not less than 35 per cent benzol

(2) The benzol shall be of such grade that not less than 80 per cent by volume shall distill below 212 deg. fahr., and its end-point shall not exceed 300 deg. fahr.

(3) The finished fuel shall have an initial boiling-point during the months of November, December, January, February and March not above 125 deg. fahr., and during the months from April 1 to October 31, not above 132 deg. fahr., and the end-point not above 420 deg. fahr.

(4) The finished fuel shall yield upon distillation not less than 85 per cent by volume at 360 deg. fahr., and not less than 95 per cent by volume at 405 deg. fahr.

(5) The sulphur content shall not be greater than 0.15 per cent.

Having in mind the continuance of our investigations on a somewhat larger scale, the contract for fuel included a specification for straight gasoline and also a provision for changing to any other kind of fuel that might be found desirable, with suitable arrangements for arriving at a price. In January, 1929, arrangements were made for a rather ambitious program of investigations to extend over a period of two years, the duration of the present contract, with the object of determining definitely the kind of fuel that would be most satisfactory and economical. The test will include standard gasolines and blends, and possibly special fuels.

Two-Year Program of Tests

Two groups of motorcoaches are under test operation at the present time. The first group consists of nine double-deck coaches operating on a route through the center of Baltimore from north to south, the north end being in the residential section, and the south end being in the congested business section. The topography is rolling, with a general down-grade from north to south. These motorcoaches are housed in a closed, but poorly heated, garage, and are operated about 25,000 coach-miles per month. They are equipped with six-cylinder engines of 77 maximum b.hp. and the conventional vented cooling-system. Certain straight gasolines produced a fair mileage per gallon but failed to give the necessary maximum power when required. These gasolines were found to have relatively equivalent initial boiling and 50-per cent points in comparison with other more satisfactory gasolines, but their end-points were in the neighborhood of 395 to 400 deg. fahr., whereas those of a higher end-point, around 415 to 425 deg. fahr., gave ample power under all conditions. In tests conducted so far, these high-

end-point gasolines have not given much trouble from crankcase-oil dilution but, as the tests progress into the winter months, more comprehensive information may be available.

The second group consists of ten 25-passenger single-deck coaches equipped with four-cylinder engines, and are operated on a city route from east to west, the east end being in the residential section and the west end in the congested business section. Here again the topography is rolling, the grades being about equal in both directions. Parking is largely restricted on this route, and the boulevard-stop system applies over about 60 per cent of the distance. However, the stops, slowing down and acceleration made necessary by receiving and discharging passengers are numerous. These motorcoaches are housed in an unheated garage and are operated about 26,000 coach-miles per month.

The four-cylinder engines develop 54 maximum b.hp. and are equipped with the Foutz cooling system. This is a closed water-cooling system which heats up quickly and operates at temperatures considerably higher than those of the conventional system and which are uniform under varying external conditions. Here again, the high-end-point gasolines have developed appreciably more power and shown no effect on dilution, undoubtedly because of the high cylinder-wall temperature.

Curves plotted from the daily operation of these tests, including also relative atmospheric humidity and temperature, show evidence of an increase in miles per gallon with an increase in humidity. Tests[2] conducted at the Bureau of Standards have shown decreased horsepower with increased humidity. No definite conclusion has been reached at this time as to the reason for the results indicated in the Baltimore tests, but it seems that under certain conditions, with the engine operating normally much under full power, an effect may be produced on the combustion mixture which tends to give better operation of the engine as a result of greater moisture content of the air.

Tests of each grade of fuel in one of the groups of motorcoaches are continued for a full month and an effort is made to keep the equipment in about uniform condition. A general inspection is made at the end of each test. Such tests may be open to criticism on the ground that there are many factors which cannot be measured. However, they have one advantage over laboratory tests in that the fuels are subjected to the conditions which have to be met in practice. An operation of 25,000 vehicle-miles over a period of one month should about average up all the various factors.

Fuels Improved by Research and Cooperation

The research which is being carried on in the laboratories of the Bureau of Standards and of the oil companies have made possible the development of qualities in fuels to meet the various conditions occurring in practice and have brought about an improvement in fuels now available, but so far nothing is available from such research which the operator can use in making his purchases.

There is no question that the engineers of the petroleum industry have improved their products. Better fuels are available now than could be obtained some years ago. Also, the very close cooperation among engineers of the petroleum and automotive industries has improved the efficiency and economy of the internal-combustion engine. Possibly some room for argument exists as to which of these two industries leads and which follows. The petroleum engineer says that the development of the fuel has followed engine design; on the other hand, the engine designer claims that the changes in the engine have followed the development of oil refining and the qualities of the fuel available from time to time. It is generally true that the manufacturer of engines is dependent on the fuel companies in that his design of manifolds, combustion-chambers and other engine parts can be substantially affected by any radical change in fuel. Engine design necessarily follows, in many of its details, in the wake of a fuel change.

As a result of this cooperation brought about with the assistance of the Bureau of Standards, oil companies are fully aware of the difficulties of engine design as affected by material changes in the fuel, and are working with the engine manufacturers in this respect. The paper by Dr. H. C. Dickinson entitled, The Cooperative Fuel Research and Its Results[3], summarizes the results that have been obtained. From this cooperative study the matters of dilution, starting, and acceleration are now fairly well understood, but further study is necessary especially to procure satisfactory control over the relationship between fuel and compression ratios.

It is perhaps of some importance to the operator for a decision to be reached as to whether engine design should or should not follow fuel changes. Seemingly the engine design should necessarily follow the fuel, because the conservation of our crude-oil supply is of National economic importance, and everything must, therefore, be subordinated to obtaining the most economical use of our crude oil. That places a great responsibility upon the petroleum engineer to keep both the automotive engineer and the operator informed as far in advance as possible of the trend in the development of fuel from crude oil whenever any possibility of change seems to be likely. The operators and the engine manufacturers are in the hands of the oil companies and cannot write specifications for fuel that will have much effect on refining processes.

The only way in which a specification can be obtained that will ensure to the operator a fuel in a competitive market that will meet his definite requirements is by the same kind of cooperation between the petroleum industry and the operators that obtains between the petroleum industry and the engine manufacturers. The petroleum industry has done considerable research work to ascertain the qualities needed to give the operating performance that is required. It can also produce means to be used, either in lieu of or supplementary to the present chemical analysis universally used, for conveniently testing or checking fuel purchased.

Some Spectacular Experiments

Perhaps the situation has not been thought of in this way, but the operator is like the innocent bystander; any change made by the engine designer or the gasoline producer leaves the operator in a quandary. Operators are, of course, trying to help themselves. Experiments and studies such as we have been conducting in Baltimore are being made by most of the large transportation companies, and some of these experiments

[2] See the S.A.E. JOURNAL, September, 1929, p. 277.

[3] See the S.A.E. JOURNAL, August, 1929, p. 171.

are more or less spectacular. For example, we have been experimenting in Baltimore for four years with an improved cooling system which not only assures adequate cooling at all loads, because its capacity for the dissipation of heat varies directly with the load on the engine, but also makes possible the operation of the engine at considerably higher cylinder-wall temperatures. This is the Foutz System[4] with which the four-cylinder engines used in the fuel tests previously mentioned are equipped.

The advantages of higher and more uniform cylinder-wall temperatures are well understood, and efforts have been made for a number of years by automotive engineers to develop a cooling system that would give these results. The system in use on our motorcoaches seems to solve this problem and, although the manufacturers have so far manifested little interest in it, they undoubtedly will realize its advantages. It is also of interest to the petroleum industry in that it should make possible the utilization of a somewhat lower grade, and therefore cheaper quality, of gasoline.

Another of the spectacular experiments has been the effort to use a so-called crude oil in the gasoline engine. For this purpose a vaporizer has been developed which ensures the vaporization of this fuel before it enters the cylinder. As this fuel probably corresponds somewhat to the gasolines of 1917 to 1919, it is very likely that its use would bring back the dilution troubles of that time. Even the use of vaporizers may not prevent this, for with such fuels dilution probably is caused, not so much by failure to vaporize in the first place, as by failure to remain in the vapor state in the cylinder. Maintenance of higher cylinder-wall temperatures by means of the cooling system referred to might solve these difficulties.

These fuel-oil tests are being made exclusively on gasoline-electric motorcoaches. Considerable difference might exist in its use on these vehicles as compared with a mechanically driven vehicle. The conditions of starting and accelerating are not the same, and an electric drive probably tends to smooth out any unevenness in power delivery by the engine, although a somewhat greater loss occurs between the engine and the rear wheels. The results, however, will make a valuable contribution to the study of automotive-vehicle operation.

A third spectacular experiment is the effort being made by a large-fleet operator to go directly to the Diesel engine, using ordinary fuel-oil. A motorcoach equipped with such an engine was driven to Atlantic City during the American Electric Railway Association convention, held there from Sept. 28 to Oct. 4, 1929. We were told that the oil consumption produced about 8 or 9 miles to the gallon and that the fuel cost but 5 cents per gallon as compared with 11 cents for gasoline. The statement was made that the use of these engines was entirely experimental and that no claims are made for it except that it holds interesting possibilities. Here, again, the cooling system in use in Baltimore might be very desirable; in fact, it is probably of even greater importance on the Diesel engine than on the gasoline engine. The advantages of the quick warming-up, uniform temperatures under all conditions, higher cylinder-wall temperatures, and the great reduction in the quantity of water, with resulting reduction in water-jacket size, would greatly simplify the use of Diesel engines on motorcoaches or trucks, and also in other applications.

I mention these several experiments because they represent efforts on the part of operators to find a way of using fuel of low grades. They may indicate to the oil companies and the engine manufacturers that the large-fleet operation will make it possible for operators to influence the development of engines and fuel, and show them the ideas at present prevailing in the operators' minds.

Method of Purchasing Fuel

In connection with the discussion of fuel specifications, consideration should be given to the method of purchasing fuel; that is, whether it should be bought in bulk and distributed by the user or purchased at tank-wagon prices and delivered by the oil companies. Considerable difference of opinion on this point exists among operators. Some large users are using one method and some the other. For bulk purchase it usually is necessary to install storage tanks and operate tank trucks for redistribution. Operators who are using this method feel that they are saving money by it. We have given careful consideration to this matter in Baltimore, but are still buying on tank-wagon delivery. If money can be saved by the bulk-purchase method we shall change to it, but it should be possible for the oil companies to do the handling more cheaply and satisfactorily than can the operator. This matter should be given consideration by the oil companies. Cooperative study should settle the question whether the transportation companies, as consumers, must go into the business of storing and handling fuel or would save money by leaving this local distribution to the oil companies, which are more experienced in such work.

Whatever way purchases are made, a more satisfactory basis for measuring the fuel should be adopted. A small variation in gallonage in tank-wagon deliveries or even tank-car deliveries may amount to a great deal in the course of a year. Some thought should be given to the use of a volume correction-factor to correct for change in temperature. The establishment of a uniform practice is very desirable. Accurate measuring methods also are important in the use of fuel by operating companies, and considerable improvement has been made in recent years by the companies furnishing measuring devices and pumps. This is important not only to the operating companies in connection with statistics pertaining to their costs and operating efficiencies, but also to the oil companies, as the satisfactory performance of any given fuel is often determined from such records. Experiments have been conducted for a period of years to develop a suitable gasoline meter for application to individual vehicles. We are trying out such meters in Baltimore, and if they prove to be satisfactory they will be of great value, not only in testing fuels, but in many other ways.

Kinds of Fuel Being Purchased

Some effort has been made to survey the various motorcoach operations to ascertain what kinds of fuel are being used and why. Replies[5] by 32 companies in all parts of the Country to the question, What kind of gasoline do you use? reveal that, with temperature conditions about the same, for operations in hilly country,

[4] See *Electric Railway Journal*, July, 1929.
[5] See *Bus Transportation*, June, 1929, p. 300

2 used straight gasoline and 1 used blended fuel; for rolling country, 14 used straight gasoline, 4 used blended fuel, 2 used high-test gasoline, and 1 used ethyl gasoline; and, for flat country, 7 used straight gasoline. Thus a total of 24 companies were using straight gasoline, 5 employed blended fuel, 2 used high-test gasoline, and 1 used ethyl gasoline. The reasons given by these companies for preferring the fuel each used showed a great variation of opinion.

A need exists for some uniform and reliable method of determining the best kind of fuel for the various operations and the equipment in use. The fact is recognized that tests of the kind now being made in Baltimore to find the most effective and economical fuel are not entirely accurate and that the results must be accepted with caution, particularly as between fuels of nearly the same characteristics. Yet, the results obtained in practice are, after all, what the operator is interested in and must, in the end, be the basis for his selection.

I suggest that frank and honest consideration be given to fuel specifications from the purchaser's viewpoint by engineering representatives of the three interests, and a specification or series of specifications be produced which the operator can accept with confidence. It would be unfortunate if the ideas obtained by the operators through these various experiments were to develop at variance with the economic, practical conditions in the petroleum and motor-vehicle manufacturing industries. Cooperation of the kind that has brought such excellent results between the petroleum and the automotive industries can and will produce equally satisfactory results between these two industries and the fleet operators. The purchasing of fuel in large quantities for fleet operation should be done in fairness to both the operator and the fuel companies. The operator wants to obtain at the lowest possible price a fuel that is best suited to his conditions or his type of engine. He expects the specification to bring in competitive bidding. There is no reason why, in an industry unregulated by public authority, a company which can produce a fuel more efficiently or economically than another should not be able to profit by that fact in submitting bids. On the other hand, each oil company wants to know, when submitting its best price for the product specified, that every other bidder will be required to supply the same grade of product he is bidding upon, and that, knowing beyond doubt that this is the case, they will submit honest bids.

What Future Specifications Must Require

The indications are that specifications for gasoline for the future shall require:

(1) Ability to give good starting at the lowest temperature to which it is subjected, to be determined by temperature at the 10-per cent point of the A.S.T.M. distillation curve and the slope of the curve at this point

(2) That at the 50-per cent point the curve shall be low enough to give satisfactory acceleration

(3) A high enough temperature at the 90-per cent point to give ample power when required.

In addition, the fuel must have low knock-value and the sulphur content must not exceed 0.10 per cent. The new Federal specifications, test methods and requirements give a fair outline of the possible methods to be used. More important, however, than this is the development of some method by which the operator can perform tests upon samples of the fuel as delivered to assure him that the fuel has the qualities specified. The operator should not be concerned with what is put into the fuel or how it is refined or blended, but rather with the results that can be obtained from it.

Some test devices are now in use to a limited extent, such as the Midgley bouncing-pin machine for determining antiknock properties, and the Ricardo machine for determining the highest useful compression. The development of test equipment and methods that can be used in common practice is very desirable.

The automotive engineer might also have a part in the production of suitable specifications. As it is claimed that engine design follows the changes in fuel as produced by the oil companies, the designer must know the relation between various grades of fuel and the engine characteristics. At present, when equipment is purchased, this information is not offered to the operator so that he can use it in providing the most suitable kind of fuel; in fact, it sometimes is difficult to obtain such information, although in the last year or so certain automobile manufacturers have definitely recommended certain grades of fuel. The aid of the manufacturer's engineer would be most helpful to the operator in preparing fuel specifications.

Four grades of fuel are generally available, as follows:

(1) Straight-run gasoline
(2) High-test gasoline
(3) Benzol-blended fuel
(4) Ethyl-blended gasoline

I should like to ask the manufacturers to make known which of these four grades they recommend to be used for each model they are now producing. This would influence the operator not to use a fuel which is markedly unsuitable for the engine.

I should like to ask the oil companies to submit a specification and methods of test for each one of these four grades of fuel. As the operator cannot determine accurately, by any kind of test or experiment, the differences between fuels of about the same grade, it will be better to eliminate any differences; therefore, I propose standardization of the grades of fuel. Perhaps we should have five or six grades, but no one can produce good reasons why they should not be standardized.

If these things are done, the operator can be sure of obtaining correct fuel for his equipment and the benefits of the lowest legitimate prices in a competitive market. The motorcoach and motor-truck manufacturers can be sure of the best performance of their equipment up to the limits of its design. The oil companies can depend upon equal and fair opportunity to capitalize their enterprise and efficiency.

This is an ideal, to be sure, but every development in our civilization is the realization of an ideal. This one can be realized through cooperative effort. The preceding papers[6] at this meeting are sufficient proof of that. Surely, no organization is better fitted to achieve such a result than our Society, and to do so without interfering in any way with continued development and progress.

[6] Gum in Gasoline, S.A.E. JOURNAL, January, 1930, p. 31. Also, Practical Significance of Fuel-Knock Ratings, by Earl Bartholomew; and Laboratory Testing of Motor Lubricants and Gasolines under Summer and Winter Conditions, by H. E. Becker and W. C. Bauer, not published.

Six-Wheel Motor-Vehicle Development History

By E. W. Templin[1]

Transportation Meeting Paper — *Illustrated with* Photographs and Drawings

CONCEPTION in 1918 of the idea of placing the dual rear wheels of motor-trucks in tandem, with the object of circumventing the insurmountable problems presented by side-by-side placement of very large pneumatic tires on heavy-duty trucks so that the trucks could be operated at the high speed that is feasible with pneumatic tires, is attributed to Paul W. Litchfield. Active development of the tandem construction was started at once by the engineers of the Goodyear Tire & Rubber Co., and the first vehicle embodying the idea was put in operation inside of 30 days. This ran 10,000 miles in the first year and then was scrapped. In May of the following year the task of designing an improved six-wheel construction was assigned to the author of this paper, who describes his first and succeeding designs, up to and including that produced by the Six Wheel Co., of which he became engineer in 1924. In particular, the development of means for resisting driving-axle torque-reaction, which presented the most difficult mechanical problem, is described.

Early skepticism of motor-truck and motorcoach designers is mentioned, but this is stated to have largely disappeared in the face of millions of miles of operation of six-wheel vehicles. Why it has been worth while to go to all the trouble of designing and building this type is explained.

The Goodyear experimental trucks are believed by the author to have been the forerunners of modern six-wheel trucks and coaches built in this Country and in Europe, for very plausible reasons set forth.

Data are given of impact tests and measurement of stresses in road pavement as made by the Bureau of Public Roads with six-wheel pneumatic-tired trucks and four-wheel trucks fitted with various types of tire.

In conclusion, the assertion is made that the six-wheel vehicle is the logical solution of the problem of hauling heavy loads economically without destroying the roads.

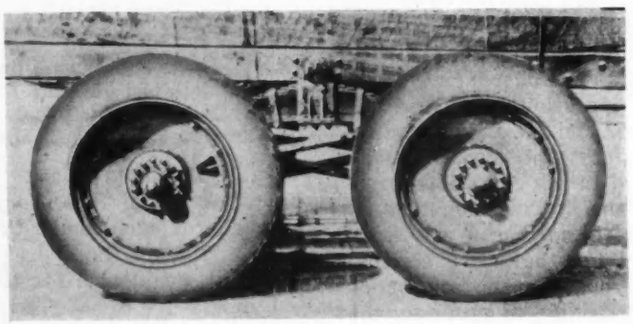


Fig. 1—First Goodyear Tandem Rear-Axle Arrangement, as Adapted to a Packard Truck in 1919

ACTIVE development and promotion of the six-wheel idea applied to motor-trucks and motorcoaches is attributed to the Goodyear Tire & Rubber Co., and particularly to the man who is now president of that company, Paul W. Litchfield. The conception of the idea by him occurred in a factory council meeting at which the insurmountable problems in connection with the operation of 48 x 12-in. heavy-duty pneumatic tires were being discussed in the year 1918. The solution suggested by Mr. Litchfield at that time was to put two smaller tires in tandem to replace the excessively large ones. How it was to be done he did not say, but he did direct the factory engineering department to proceed at once with the construction of such a vehicle. I was told that inside of 30 days the vehicle was on the street in operation. This truck, although crude in some respects, ran for a year, or 10,000 miles, before it was scrapped.

In May, 1919, the task of designing a new six-wheel vehicle was assigned to me, with the object in view of making a truck with this arrangement of rear wheels that would be satisfactory in operation and demonstrate to the motor-vehicle industry that such a vehicle was feasible.

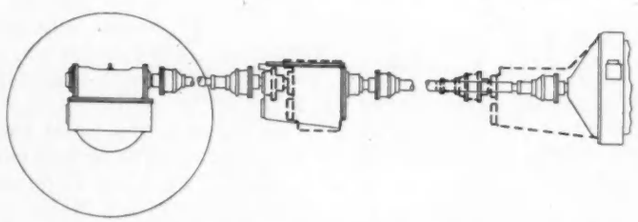


Fig. 2—Primary Three-Speed and Auxiliary Two-Speed Transmissions Used in Early 5-Ton Six-Wheel Trucks Built by Goodyear

The Auxiliary Transmission Enabled the Axles To Be Geared To Give a Speed of 40 M.P.H. This Combination Type of Transmission Is Now Coming into Somewhat Common Use To Give High Drive-Wheel Speed

Our first undertaking was the adaptation of a tandem rear-axle arrangement, as shown in Fig. 1, to the rear of a Packard truck. The construction was made up of two comparatively small worm-drive rear axles spaced apart by the rear springs, which were inverted from the usual position; and the axles were kept from turning by a torque system consisting of diagonal rods swiveled at the ends.

[1] M.S.A.E.—Formerly motor-truck engineer, Goodyear Tire & Rubber Co., Akron, Ohio.

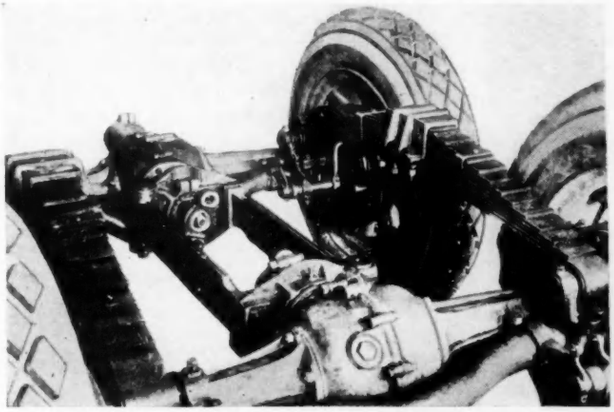


FIG. 3—ASSEMBLY OF INTERNAL-GEAR REAR-AXLES IN GOODYEAR 5-TON TRUCKS

The Forward Axle Took All the Power Through the Pinion and Ring-Gear and Transmitted Part of the Power to the Rearmost Axle. The Torque-Arms Were Made of Spring Leaves Set on Edge, Hinged at One End and Free To Slide at the Other End

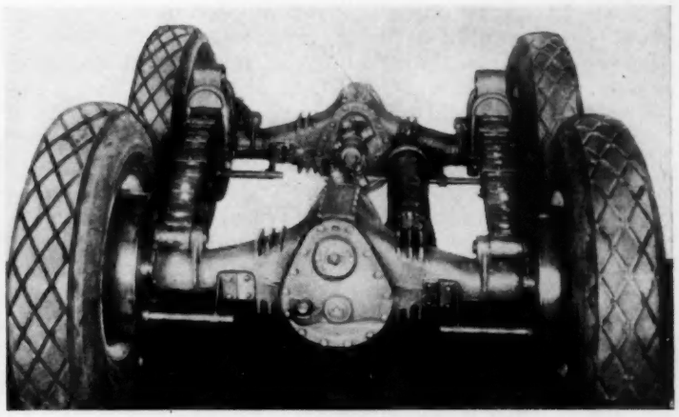


FIG. 4—TELESCOPING TUBULAR TORQUE-SYSTEM USED IN EXPERIMENTAL TRUCKS BUILT IN 1919

The Inside of the Torque-Arm Was Filled with Heavy Oil Which Lubricated the Sliding Surfaces and the Hinges at the Ends. This System Proved so Satisfactory that Its Use Was Continued in Use in Subsequent Constructions

This system worked satisfactorily for several thousand miles, including trips from Akron, Ohio, to Boston and return. It finally caused trouble from the fact that this arrangement of torque-rods causes all braking and driving torque within the axle to be taken at one end, and axle housings are not ordinarily strong enough to withstand this high stress. However, the performance of this job was encouraging, and the next step in the program was to build two complete 5-ton trucks, one employing worm-drive axles and the other internal-gear-drive axles, as the truck industry at that time, the latter part of 1919, was about equally divided in opinion regarding the merits of these two types of axle. Other main features of these trucks were:

(1) Enough engine power to drive a 5-ton truck at a speed of 40 m.p.h.

(2) Axles geared to give that speed

(3) Transmissions geared down to compensate for the gearing up of the rear axles.

The main object of these specifications, which were unusual at that time, was to make possible the operation of these trucks at the speed that it was then evident could be got with pneumatic truck-tires without causing injury to the chassis mechanism.

The trucks were designed from the ground up to meet the new conditions and yet be capable of giving continuous satisfactory service. They were, however, built from units procurable in the market. The power was provided by a 5 x 6-in. four-cylinder engine similar in design to the Class-B truck engine. The transmission was a three-speed unit-powerplant job with a two-speed auxiliary mounted amidships, as in Fig. 2. This type of transmission is now coming into somewhat general use to accomplish the same results as were desired in those Goodyear jobs of 10 years ago.

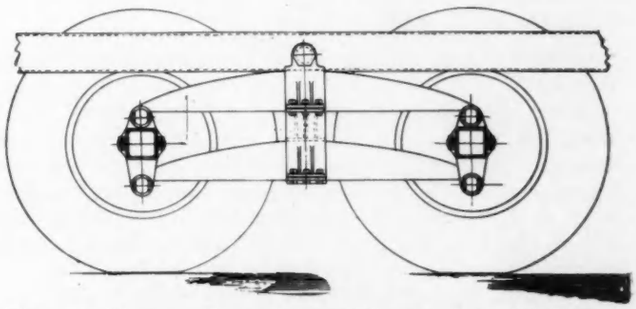


FIG. 6—TANDEM-AXLE SPRING-SUSPENSION SIX-WHEEL 44-PASSENGER MOTORCOACH

This Type of Suspension, Which Served also as a Torque-Resisting Means, Is Now Used Generally in Six-Wheel Trucks Built in England

The worm-drive axles were hooked up the same way as is shown in Fig. 1. The internal-gear axles were assembled as shown in Fig. 3. The forward rear axle took all the power through the pinion and ring-gear and trans-

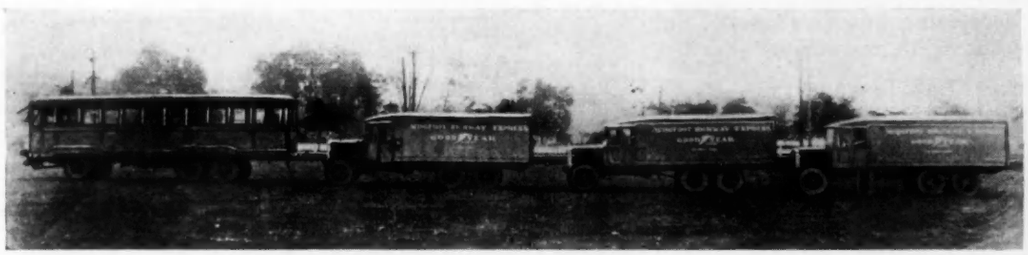


FIG. 5—LINE-UP OF SIX-WHEEL VEHICLES BUILT TO VARIOUS SPECIFICATIONS IN THE GOODYEAR DEVELOPMENT

The Leading Vehicle Is a 44-Passenger Motorcoach Built in 1920 and Equipped with Pneumatic Tires. The Tandem-Axle Spring-Suspension Is Shown in Fig. 6. The Last Truck in the String Made a Demonstration Transcontinental Trip of Nearly 4000 Miles in October, 1921, in 6½ Days' Running Time. The Six-Wheel Trucks Were Used by the Bureau of Public Roads To Secure Data on Highway-Pavement Stresses, Subsoil Pressures and Magnitude of Tire Impacts on Pavements

mitted the power for the rear axle through a second pinion meshed with the ring-gear at the back. Torque was taken through torque-arms made of spring leaves set on edge and hinged at one end and free to slide at the other end, one mounting swiveling about the third-member housing. This arrangement was abandoned very quickly, owing principally to the excessive noise of the complicated gearing.

It may be noted here that the recent development of the hypoid gear would make possible the application of axles of the internal-gear or a similar type to the

FIG. 7—CLASS-B TRUCK CONVERTED INTO A SIX-WHEELER BY THE MOTOR TRANSPORT DIVISION OF THE ARMY QUARTERMASTER CORPS

six-wheeler and avoidance of the complication mentioned.

After discontinuing the diagonal torque-rods used with the worm-drive axles, a somewhat conventional type of double torque-arm was tried. This system consisted simply of two torque-arms hinged to the two axles in a conventional way, but interconnected by the use of shackles, the pins of which bore spherical parts carried in spring-cushioned spherical seats. This system worked very well but the spherical parts wore abnormally and consequently became very noisy.

A subsequent development was a telescoping tubular torque-system, shown in Fig. 4. This consisted of two tubular members hinged to the two axle-housings in the same way as the two torque-arms just mentioned, one of the tubes being a working fit within the other. The space inside was then filled with heavy oil, which served to lubricate the hinge joints as well as the sliding surfaces. This torque system was quite satisfactory and was continued in use.

Many other details required development and improvement in the course of developing the six-wheelers, but the torque system was the main mechanical problem.

Early Skepticism Has Disappeared

Early skepticism regarding the six-wheeler was expressed in the questions:

(1) How does it turn corners?
(2) How does it run without a differential between the axles?
(3) Why go to so much trouble?

The first of these doubts has now disappeared after millions of miles of experience. However, an early conclusion arrived at by the Goodyear experimenters was that the turning of corners was a minor cause for worry in connection with six-wheel operation. The second has

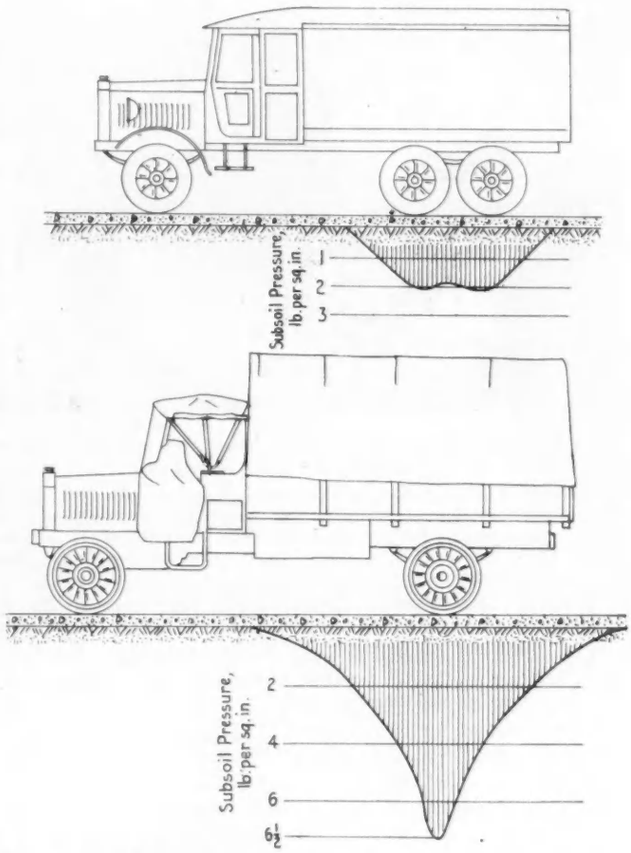


FIG. 8—PNEUMATIC-TIRED SIX-WHEEL TRUCK AND SOLID-TIRED FOUR-WHEEL ARMY TRUCK USED IN SUBSOIL-PRESSURE TESTS

The Load on Each Truck Was 5 Tons

similarly disappeared and, so far as I know, no six-wheelers are being built either here or in Europe that employ the differential between axles. A few designers started out with them and later discontinued them; namely, the Büssing Co., of Germany, and the Associated Equipment Co., of England.

As for the third question, Why go to so much trouble in building a truck?, Mr. Litchfield's vision was that pneumatic truck-tires were the right kind for trucks and, from experience, that the section size would be

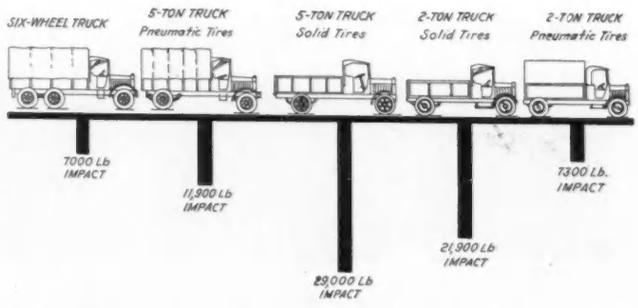


FIG. 9—COMPARISON OF ROAD IMPACT BY REAR WHEELS OF FOUR-WHEEL AND SIX-WHEEL TRUCKS EQUIPPED WITH SOLID AND PNEUMATIC TIRES

From Data Determined by Tests Made by the Bureau of Public Roads

limited to a maximum of 9 in.; therefore, to equip large trucks with pneumatic tires presented a problem worth solving, so as to make heavy trucks capable of high speed and of negotiating bad places. Another factor, not generally considered important 10 years ago, was the effect of heavy trucks upon the highways. These reasons were amply important to make it worth while to go to some trouble in the redesigning of trucks to meet the conditions. During the last year these factors have become generally realized, and this has been the main impetus behind the general favorable consideration of the six-wheeler as a means of hauling heavy loads economically and without destruction of highways.

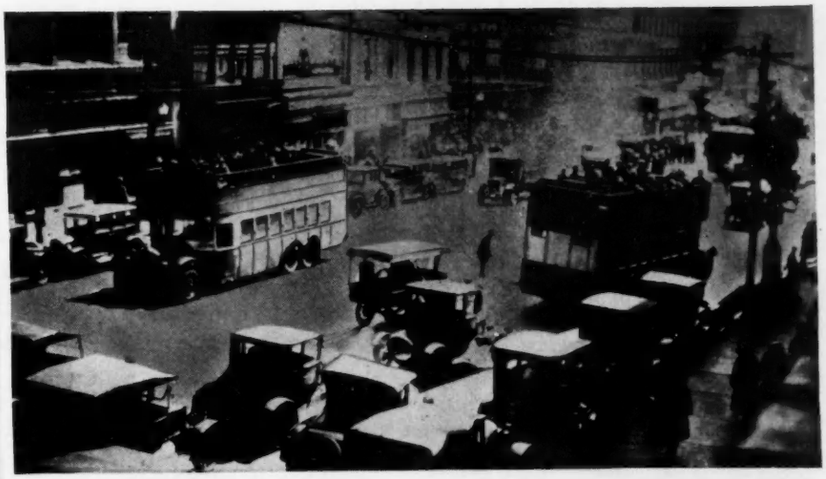


FIG. 10—FLEET OF SINGLE AND DOUBLE-DECK MOTORCOACHES OPERATING IN DETROIT

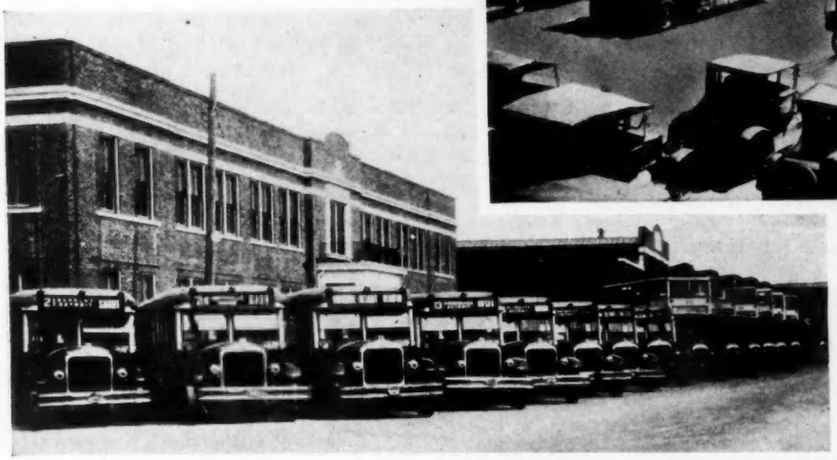


FIG. 11—SIX-WHEEL DOUBLE-DECK MOTORCOACHES IN METROPOLITAN SERVICE IN THE UNITED STATES

Forerunners of Modern Six-Wheel Vehicles

To refer again to the Goodyear development, a number of vehicles were built to various specifications. A group of these vehicles is shown in Fig. 5. The forward vehicle is a 44-passenger motorcoach built in 1920 to demonstrate that large numbers of passengers could be carried on pneumatic tires. It is interesting to relate that Frank Fageol took a keen interest in reviewing the drawings of this coach 10 years ago; and I believe the idea remained in his mind until he began building the modern Twin Coach, which, by the way, is a much better job than its forerunner. This Goodyear coach was equipped with a tandem-axle spring-suspension, as shown in Fig. 6, and this served as a torque-resisting means as well. Incidentally, this type of suspension and torque system is used generally in six-wheel trucks built in England.

The last truck shown in Fig. 5 is one that the Goodyear company considered well enough perfected to try for a transcontinental run. The previous record for a heavy truck was held by a Goodyear-owned Packard four-wheel truck equipped with pneumatics, which made the trip of approximately 4000 miles in 13½ days of running time in 1920. The six-wheel truck made the run in October, 1921, in the running time of 6½ days, or less than one-half of the time and at approximately twice the average speed. This performance was regarded as a very conclusive demonstration that the six-wheel truck had been brought to a satisfactory stage of development.

In 1922 the Motor Transport Division of the Quartermaster Corps became interested in the six-wheel construction and arranged with the Goodyear company to

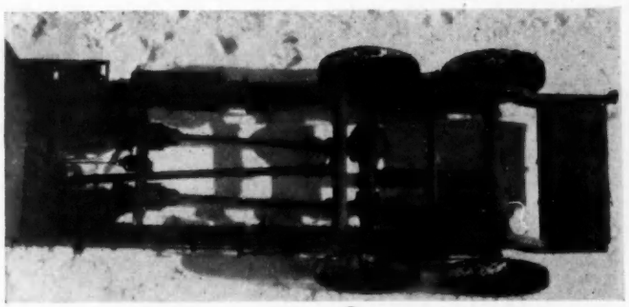


FIG. 12—APPLICATION OF GASOLINE-ELECTRIC DRIVE TO SIX-WHEEL CONSTRUCTION

Each of the Rear Axles Is Driven Independently by One of the Electric Motors

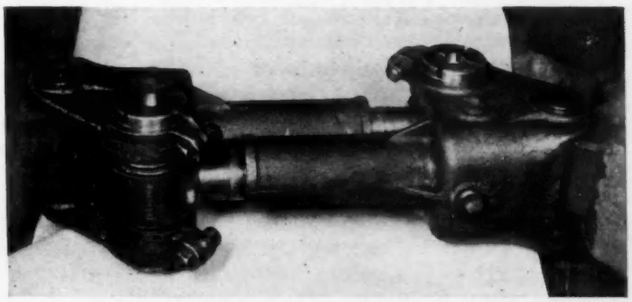


FIG. 13—LATEST DEVELOPMENT OF TORQUE SYSTEM BY THE SIX WHEEL CO.

This System Was Designed to Retain the Good Features of the Telescoping-Tube System Shown in Fig. 4 and, in Addition, To Slightly Cushion the Torque Reactions. It Was Applied to about 80 Six-Wheel Motorcoaches and Is Proving Very Satisfactory

convert a Class-B truck into a six-wheeler. This truck as completed is shown in Fig. 7. It served to prove to the Motor Transport Division the many advantages of the six-wheeler in military operations.

This truck and the Goodyear six-wheelers were used by the Bureau of Public Roads in securing data with reference to the avoidance of destructive stresses in highway pavement by the use of such trucks. Fig. 8 presents a comparison of the subsoil pressures under the rear wheels of a conventional 5-ton truck and a six-wheel truck of the same capacity from data taken by the Bureau of Public Roads. Fig. 9 gives the data taken by the Bureau in impact tests on the highway with a six-wheel truck fitted with pneumatic tires and four-wheel trucks fitted with solid and pneumatic tires. They show conclusively that the six-wheeler of heavy capacity is much less destructive of the highway than the conventional heavy truck with solid tires.

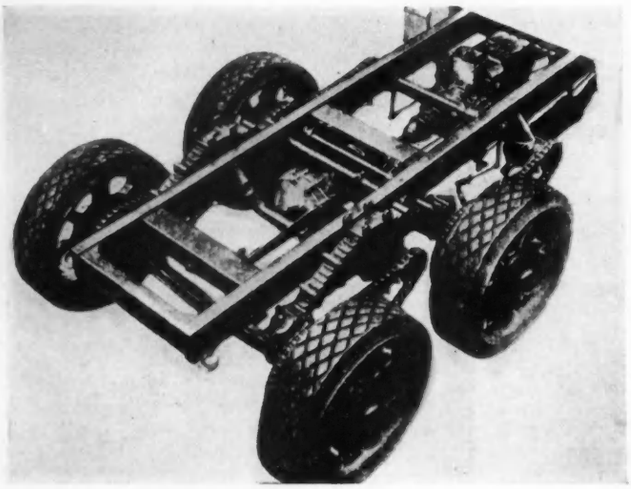


FIG. 14—MACK SIX-WHEEL CHAIN-DRIVE-TRUCK CONSTRUCTION

Hypoid Gears Are Used in the Two Jack-Shafts and the Power Shaft Passes Through the Forward Gear-Housing above the Jack-Shaft to the Second Jack-Shaft. The Springs Are Inverted and Torque-Arms Are Used as in the Conventional Truck. For Details of Construction, See Plan Drawing in the S.A.E. JOURNAL for December, p. 592

The Idea Introduced in Europe

An important event in the development and promotion of six-wheel vehicles was a trip to England and the Continent made in the winter of 1922 and 1923, by C. M. McCreery, who was at that time, and in fact throughout, in active charge of the development and promotion of the six-wheel vehicle for Goodyear. This trip was made for the distinct purpose of interesting foreign motor-truck and motorcoach builders in the manufacture of six-wheel vehicles. I confidently believe that his missionary work caused the general adoption of the six-wheeler there. The first company to build them in Europe was the Büssing Co., of Germany. One of its six-wheel motorcoaches was exhibited at the American Electric Railway Association convention in Atlantic City, in October, 1929.

So far, the largest number of six-wheelers built in this Country by any one concern were those built by the Six Wheel Co., of Philadelphia, which is not now in active operation. It built about 500 motorcoaches and trucks of this type. The principal operations of these coaches are in Kansas City, Mo.; Detroit, Cleveland, and New York City, in the United States; and from Damascus to Bagdad in Syria. (See Figs. 10 and 11.)

The performance of the vehicles in all of these operations continues to be quite satisfactory, I understand, and the jobs in operation in Syria are reported to have out-performed and stood up better than European makes in the same service.

Recent Development

A gasoline-electric application of the six-wheel construction is shown in Fig. 12. Each of the rear axles is driven independently by one of the motors.

The Six Wheel Co.'s latest development in the torque system is shown by Fig. 13. This was designed to retain all of the good features of the telescoping-tube system and yet provide a slight amount of cushioning of the torque reactions. The system was applied to about 80 six-wheel motorcoaches in operation in the Bronx, New York City, and is working out very satisfactorily.

Each truck manufacturer, as he became interested in building six-wheel vehicles, naturally desired to build them as he thought they should be built; consequently we now have various ways of accomplishing the objects sought. Some of these look to me as if they will be successful, while others will require considerable development work to perfect. The interesting fact to me is that the general inclination is to try to depart from the Goodyear design so as to avoid paying the very nominal royalty asked for the use of it; but in most cases the new designs are enough more expensive to pay the royalty three to four times.

One design on the market that has the appearance of being very good is that built by the International Motor Co. This design, shown in Fig. 14, involves two jack-shafts using hypoid gears so that the power shaft can pass through the first gear housing to the second shaft. The jack-shafts then drive chains in the usual way, and torque-arms also are used the same way as in the conventional truck, the springs being mounted inverted, as in the Goodyear designs.

The six-wheel vehicle is the logical solution of the problem of hauling heavy loads economically without causing destruction of the highways.

Discussion of Six-Wheel-Vehicle Papers[1]

Interest Centered on Differentials, Conversion of Four-Wheelers to Six-Wheelers, Tire Sizes and Capacities, and Trailer Operation

WHETHER tires of greater carrying capacity can be provided by the tire manufacturers is a question debated first, and it includes considerations of the effects of torque reaction on tires. Dual rear-tires for six-wheel vehicles are said to be most advantageous. The effects of a certain type of gasoline-electric six-wheel motorcoach for city service in eliminating some of the problems relating to universal-joints, differentials and braking are explained. Diagrams showing how weight shifts between rear bogie-unit axles are analyzed. The conversion of four-wheel into six-wheel vehicles by applying a secondary axle is another subject of debate, together with whether the use of a differential between the driving axles is essential. Other questions raised and discussed include tractor-trailer weight-distribution and the haulage of long-distance freight, details relating to tire sizes, over-all widths, difficulties experienced with differentials, and general considerations on the subject of the dangers of converting vehicles.

A. J. BROSSEAU[2]:—Regarding the carrying capacity of a pneumatic tire, is there any hope, based on the tire manufacturers' experience, that they can provide a tire of greater carrying capacity or a longer life for the tires which we are now using? I am told that the tire manufacturers rate the 34 x 7-in. tire for motor-truck or motorcoach use at about 3000-lb. capacity. Can we hope to get a 34 x 7-in. tire for such service that will give reasonably good performance and carry 4000 lb.?

G. M. SPROWLS[3]:—Whenever the load is increased beyond the rated carrying-capacity, the mileage decreases very rapidly, not only because of tread wear but also on account of blow-outs. During the last several years the mileage given by truck tires has been greatly increased; if a tire does not give at least 18,000 to 20,000 miles of service, the user is not satisfied. It is the amount of air which the tire carries that determines the carrying capacity, and the volume of air cannot be increased without increasing the size of the tire. I cannot hold out much hope that the carrying capacity for a given size of tire will be increased unless the user will be satisfied with lower mileages than he is now obtaining.

J. W. SHIELDS[4]:—We all recognize that tires are a main reason for the development of a six-wheel unit, because we must have more tires to carry the load. Regarding future tire capacity, we hope that we can make tires so that they will carry greater loads for greater distances; but it depends entirely on what will satisfy the operator in the matter of distance. The 34 x 7-in. tire will today carry a 4000-lb. load, but not so far as it will carry a 3000-lb. load. The tire engineer knows that the mileage or service we can expect from a tire varies inversely as the square of the load it carries; hence, if the load is increased, the service obtained from the tire is decreased very rapidly. We have just completed development work on a pneumatic tire of approximately 13½-in. cross-section diameter which is rated to carry 7700 lb., at the special request of Mr. Brosseau's company.

We have had a great amount of experience with tires on six-wheel units; some of it has been extremely satisfactory and some of it very unsatisfactory. It is difficult to draw a very sharp distinction between the reasons that might be pointed out for the difference in tire service, but we have found that in many operations the lack of a differential has been a serious cause of tire trouble. A differential between the rear axles is used on one particular operation and the tires have given most wonderful service, although the service is extremely severe.

Torque-Reaction Effects on Tires

When power is applied to the six-wheel unit a torque reaction is set up which tends to raise the middle axle and increase the load on the rear axle. Based on calculations I have made on one of the well-known six-wheel units that is in operation, we can expect that, when operating under what we might call a normal pull on a level road, the difference in the load distribution produced by the engine torque will cause a distribution of the load which will change the deflection of the tires on the two rear-axles to create as much as ½ to ¾ in. difference between the rolling radii of the two tires. In an operation where no differential is used between the two axles, even though the tires may be of exactly the same dimensions and inflated to carry exactly the same load, the tires must undergo a slipping action at every revolution equivalent to more than six times the difference between the rolling radii. In some cases in which brand-new tires of exactly the same size are installed on a unit and inflated as nearly as possible to equal pressures, there is a tendency for one tire to assume a majority if not all of the slippage, and that one tire will wear very rapidly. I have seen cases in which a new tire has been completely ruined in less than

[1] Presented at the Transportation Meeting, Toronto, Canada, by E. W. Templin, see p. 54; and by L. R. Buckendale, A. M. Wolf and G. M. Sprowls, the last three papers having been printed in the S. A. E. JOURNAL, December, 1929, beginning on pp. 583, 589 and 600, respectively.

[2] A.S.A.E.—President, International Motor Co., New York City.

[3] M.S.A.E.—Manager, highway transportation department, Goodyear Tire & Rubber Co., Akron, Ohio.

[4] M.S.A.E.—Sales engineer, Firestone Tire & Rubber Co., Akron, Ohio.

1000 miles, when the three other tires on the rear end of the unit were still in excellent condition and showed no indication of slippage. Exactly the reverse of this action occurs when the brakes are applied, and the load is then thrown on the front axle.

I think the industry should give very serious consideration to the action of the differential, if both axles are driven, but is it absolutely necessary that both rear axles be driven? We think nothing of hooking one or two trailers behind a standard four-wheel truck and letting one axle do all the driving. Why should it be more essential for the same load to have all the driving distributed between the two axles? Some cases exist in which only one axle on a six-wheel unit is used to drive, but I believe that in the majority of such cases the very serious error has been made of driving on the front axle, because the torque reaction from the engine tends to raise that front axle and reduce the tractive effort as soon as power is applied. If that driving effort were applied on the rear axle so that the torque effect would tend to increase the tractive effort, I believe that in the majority of operations on smooth roads a single-axle drive would be sufficient in case no trailers are hauled.

Dual Rear-Tires Advantageous

Regarding single versus dual tires, the six-wheel truck was at least first conceived to eliminate dual-tire equipment. But dual tires have a very great advantage over single tires on either a four-wheel or a six-wheel vehicle. I believe that the future six-wheel vehicle will have dual-tire equipment. I shall quote prices on tires that would be required to equip a job with single-tire equipment or corresponding dual-tire equipment of the same capacity. Assume a 42 x 9-in. single tire which has a carrying capacity of 5000 lb. and compare it with two 36 x 6-in. tires each having a capacity of 2500 lb., a total for the two of 5000 lb. and equal to the capacity of the 42 x 9-in. tire. One 42 x 9-in. tire as quoted today in the United States costs $169.30; and the two 36 x 6-in. tires cost $103.60. Hence, by using dual tires, one can secure the same tire capacity at about 60 per cent of the cost; and, so far as service is concerned, our experience indicates that one can expect considerably better mileage and less tire trouble from the dual equipment than from the single equipment.

Gasoline-Electric Bogie Provides Differential Action

THOMAS S. KEMBLE[5]:—Mr. Wolf has classified and illustrated certain actual and possible combinations of structure for vehicles on six or more wheels and has called special attention to the possibilities of extreme angularity of universal-joints, of tire-fighting in the absence of inter-axle differentials, of ineffectiveness of drive when one wheel slips where a conventional inter-axle differential is used, and to variations of axle static-loads arising from certain methods of absorbing axle torque-reaction. Some description of a gasoline-electric six-wheel vehicle which meets a few of these problems from a different angle is therefore of interest in connection with his paper.

Fig. 1 shows a motorcoach of this construction as built by the St. Louis Car Co., and Figs. 2 and 3 show the bogie unit in which all problems arising from universal-joints, differentials, tire-fighting and loss of traction due to differentials, are eliminated by supplying individual motor drive for each of the bogie wheels, the motors being incorporated in the axles parallel to the wheel centers. This construction would involve impractical unsprung-axle weight if the motors were located anywhere in a vertical plane through the wheel centers; but these motors are offset in a horizontal direction from the wheel centers toward the torque support, thus reducing the effective unsprung weight to a value even less than that of a conventional axle of corresponding capacity. By this construction and using the motor frames and the gear housings as the conventional load-carrying axle-casing, a net saving of about 1000 lb. is effected in this vehicle.

The individual motor drive for each wheel constitutes a perfect differential in that each of the four motors can run a little faster or a little slower than the others, and at the same time the effect of one or more spinning wheels in reducing the power of the other wheels is eliminated as well as the damaging effect on the tire from wheels spinning at high speed and then suddenly taking hold.

At the beginning of this development, interrelated problems of spring suspension, torque reaction and transfer of static loads on the axles together with such factors as simplicity, weight and cost, were analyzed most carefully. Various possible arrangements of spring suspension possess certain theoretical advantages, but the arrangement of a single spring at each side of the chassis, pivoted at the center and connected to the axles by the conventional spring-eye pin or its equivalent seems to be the simplest and lightest and to include all the flexibility desired, together with minimum unsprung-weight and cost.

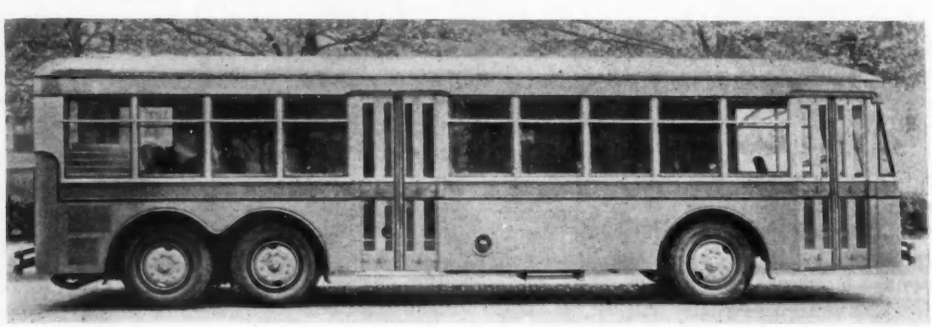


FIG. 1—GASOLINE-ELECTRIC SIX-WHEEL MOTORCOACH BUILT BY THE ST. LOUIS CAR CO.

Objection to concentration of load on the chassis side-frame where the spring pivot is attached seems to be purely theoretical in view of the ease and low cost of reinforcing the side frame locally to take care of this concentration. Objections to transfer of part of the load from one axle to the other were given careful consideration. These axles could be arranged with each torque support forward of its axle, or to the rear, or with the torque support of the rearward axle to the

[5] M.S.A.E.—Consulting engineer, St. Louis.

rear and that of the intermediate axle forward, or with both torque supports between the axles. The last arrangement is the most simple, affords a quickly replaceable bogie unit and involves the least weight and cost, but if the spring eyes and spring pivots were to be located high off the road, as indicated in Fig. 12, Type I, of Mr. Wolf's paper, the load transfer from one axle to the other would become a serious matter. By arranging the spring eyes and pivots well below the wheel centers, as in the bogie illustrated in Fig. 2, the load transfer is so reduced that practical difficulty is eliminated while the advantages of simplicity and quick replacability with minimum weight and cost are retained.

This combination has been used successfully over long stretches of mud road in Missouri bottom-land which were entirely impassable for conventional four-wheel vehicles. It also has operated successfully over wet high-crowned clay roads and "soapstone" hills in Missouri at times when all other traffic had been forced to abandon operation. It will not, of course, operate successfully over such terrain as requires individual drive of all wheels. However, it is believed that, outside of a very limited field, neither the reversing of the intermediate axle with its torque support nor the addition of driving motors for the front wheels would be economically justified.

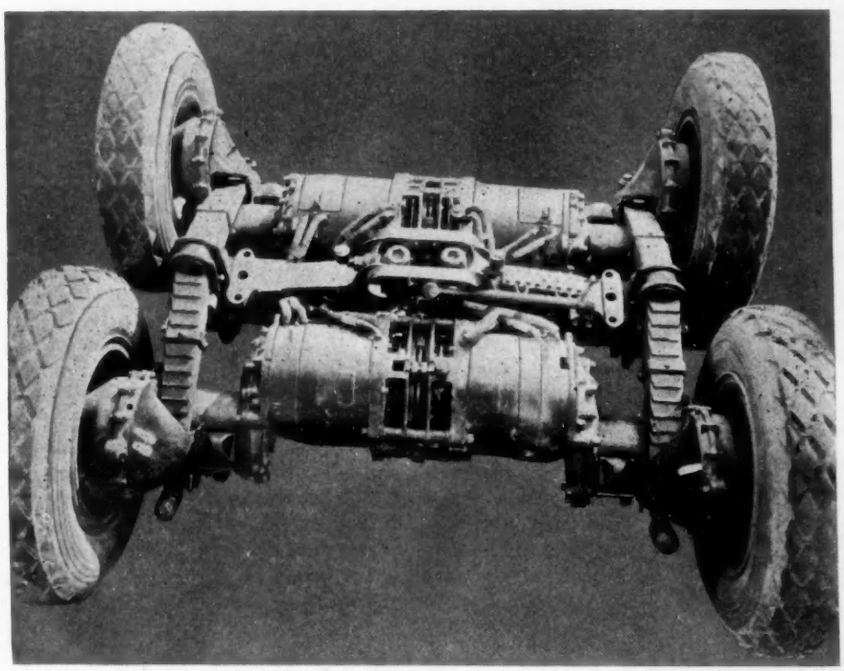


FIG. 2—REAR BOGIE-UNIT OF THE SIX-WHEEL GASOLINE-ELECTRIC CITY-SERVICE MOTORCOACH SHOWN IN FIG. 1

The Motors Supply Individual Drive for Each of the Wheels, Being Incorporated in the Axles Parallel to the Wheel Centers, Thus Eliminating the Problems Arising from the Use of Universal-Joints and Differentials

Regenerative Electric Braking

Regenerative electric braking on four wheels is employed for coasting down grade and for all service stops at speeds above 5 m.p.h. Air-cooled mechanical brakes at the center of each axle are used for parking and to complete the stop after the speed has been reduced to 5 m.p.h. by the electrical brake. The mechanical brakes are arranged so that the use of auxiliary power for their operation would be superfluous. Brake-shoe wear is reduced to almost negligible proportions, expense and maintenance of auxiliary power equipment for brake operation is eliminated, and all braking heat is completely removed from the tires.

H. A. SOULIS[6]:—What effect do the various six-wheel constructions have on scuffing and tracking of the rear tires?

A. M. WOLF[7]:—That was discussed at the 1928 Transportation Meeting in Newark, N. J., at which Ethelbert Favary presented figures[8] showing that, in making a turn, assuming that single tires are used all around on the rear bogie-unit, the relative difference between the arcs described by the two tires on one side is less than the difference in the arcs described by dual tires on the rear axle of a conventional truck. If Mr. Soulis refers to the subject of torque reaction, the equalizing member is a spring or rigid beam. It really would make little difference which it is, because, if it is subjected to torque reaction, the spring can be wound up to a certain point and then it can be considered as a rigid beam.

Mr. Shields mentioned the transfer of weight from the forward bogie-unit axle to the rear one. This is true in the Goodyear construction and was alluded to in my paper as the construction in which the torque reaction is wholly self-contained within a bogie unit. But the Timken and Wisconsin constructions throw the weight from the rear bogie-unit axle to the front one, and Figs. 4 to 7 illustrate the reasons.

The British War Office design, in which two axles are connected by parallel springs between them to which the frame load is transferred through a pivot *a*, as shown in Fig. 4, is taken as a convenient example. For purposes of discussion, the two springs can be replaced with their equivalent, considering this as one beam between the axles and that it oscillates on the pivot point. That would be similar to such a construction as the Timken equalizing beam. Then we have the axle housings and, from points on top of the worm housings *b* and *c*, a ball-jointed rod extends to the central tubular cross-member of the frame as shown at *d* in Fig. 5.

Assuming that the front of the vehicle is to the left, if the vehicle attempts to go forward, the wheel rotation is according to arrows *e* and the torque reaction of the housings is as shown by arrows *f*. Rod *g* will be in compression, rod *h* will be in tension and the tendency would be, if we consider that the tires are rigid and have no farther to deflect, to throw the center-lines through *b* and *c* over at an angle into positions *i* and *j*. In doing that it represents an effort to raise the front

[6] Chief engineer, Fruehauf Trailer Co., Detroit.
[7] M.S.A.E.—Automotive consulting engineer, Newark, N. J.
[8] See S.A.E. JOURNAL, December, 1928, p. 608.

end of the frame. The cross-member indicated at *d* is moved over to position *k*, with *a* as the center of rotation, so that the center-line through *k* parallels those indicated by *i* and *j*, which brings about this effect. The amount of movement is of course exaggerated, being shown in this way solely for clarity.

Looking at the problem in another way and considering *d* a stationary point, we have the point *b* in the upper part of Fig. 5 on the top of the housing, through which passes the torque-reaction thrust indicated by arrow *f*, and induce counterclockwise rotation; in other words, we are now trying to rotate the bogie unit around the center *a* as a pivot. If the distance *bl* is scaled off to represent graphically the torque-reaction thrust, the downward thrust is represented by *lm*, with *m* lying on the radial line *ba*. The reaction through point *c* results in an equal but upward thrust, again resulting in counterclockwise rotation. The result is that weight relieved from the rear bogie-axle is transferred to the front one.

The same action occurs in the Timken construction, the only difference being that, instead of acting against the frame, the reactions come up to the two points or ball-joints on the spring-pad casting and we get that much resiliency or "give" from the spring. The effect still is the down-thrust at the front.

The same down-thrust results also in the Wisconsin axle hook-up. The equalizing beam is located just below the center of the axles and the weight is transferred to the beam center through the spring, as indicated at the top in Fig. 6. Radius-rods *n* at either side are paralleled by a single member *o* to complete the parallelogram. A rod *p* extends directly between anchorages *b* and *c* on the worm housings. Rod *o* extends from *b* to a point *q* on a frame cross-member.

A somewhat similar general type is also shown in Fig. 6 of Mr. Buckendale's paper. We have a point of rotation similar to *a* at the center of the beam, and get the same effect as that in the upper part of Fig. 5; that is, an effort to lift the frame. Therefore, taking the reaction from the stationary point *q* on the frame, in Fig. 6, rod *o* will be in tension and we can develop, as in the lower part of Fig. 6, a diagram comparable with that in the lower part of Fig. 5, with *a* as the center of rotation. The direction of rotation is again the same.

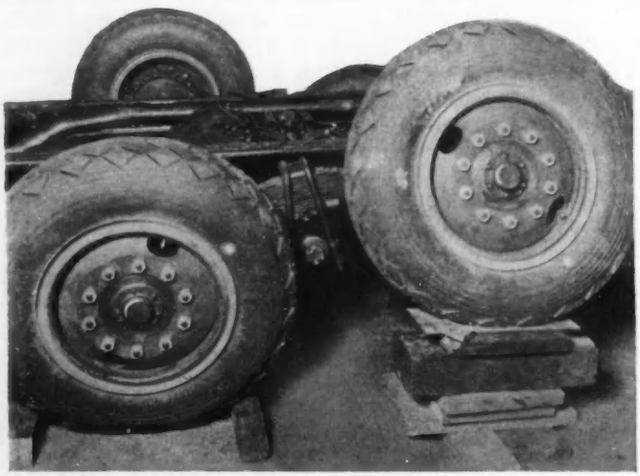


FIG. 3—DEMONSTRATION OF FLEXIBILITY OF BOGIE-UNIT SHOWN IN FIG. 2

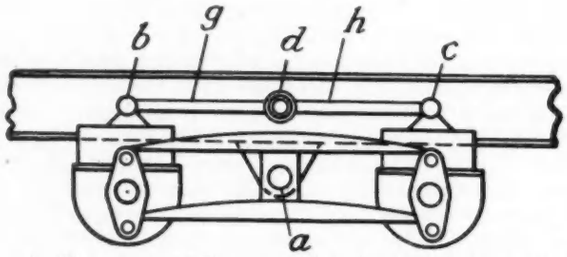


FIG. 4—DIAGRAM OF BRITISH WAR OFFICE DESIGN OF THE REAR OF A SIX-WHEEL VEHICLE

The Two Axles Are Connected by Parallel Springs to Which the Frame Load Is Transferred Through a Pivot

The reaction at *c* in the upper part of Fig. 6 is transferred to *b* through rod *p*. The down-thrust component can be resolved as before.

So the matter of the reaction is quite vital; I think many of us have not given it the amount of study it deserves. Mr. Shields certainly bears that out in his statements regarding experience with tires. He commented about the torque reaction in which only one axle drives. It is unnecessary to establish a torque connection between a live axle and a dead one. The live axle can have its own torque hook-up to the frame, and the dead axle simply carries the load through the equalizing beam or spring; therefore, this is not essential, as I see it. I have alluded only to torque reaction in these comments. Radius-rods at an angle induce vertical components under driving thrust from the axle or axles and also should be given consideration.

In Mr. Sprowls' paper, regarding the number of tire changes per mile, what type of construction was used in the vehicles under consideration? It seemed that if the rear-axle tires gave poorer service, the type of bogie unit would be of the self-contained torque-reaction type instead of the type in which the torque reaction is absorbed by the frame.

In the Goodyear construction shown in Fig. 7, the

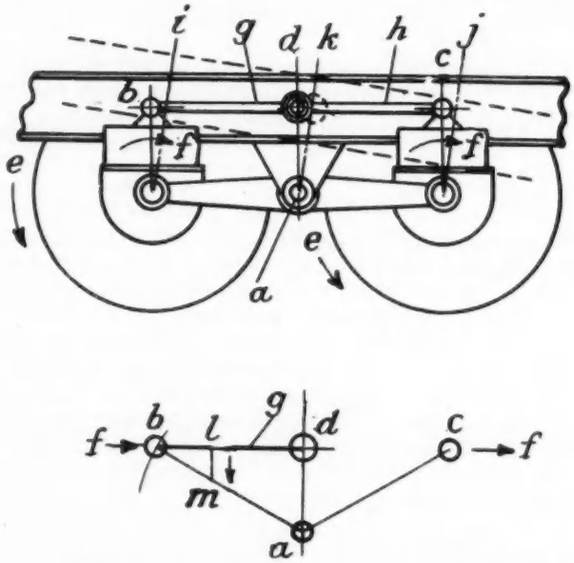


FIG. 5—DIAGRAMS SHOWING DIRECTION OF TORQUE REACTION

Considering the Front of the Vehicle To Be at the Left, If the Vehicle Attempts to Go Forward, the Wheel Rotation Is According to Arrows *e* and the Torque Reaction of the Housings Is as Shown by Arrows *f*

torque reaction of the rear axle works directly through the lever-arm r, which is equal to the distance between the centers of the axles; the force acts to raise the front axle. Similarly, the reaction of the front axle tends to depress the rear one. The self-contained unit tries to make the rear end go down, whereas the other tries to make the forward end go down.

MR. SPROWLS:—The chart shown as Fig. 1 in my paper related to some of the early six-wheel vehicles in which the rear axle took the torque reaction.

Conversion from Four-Wheels to Six

J. F. WINCHESTER[9]:—By applying a secondary axle to a 5-ton truck, can that unit be converted immediately into a 10-ton truck, provided the only other change made is to install tires of sufficient capacity to carry the load? My thought in asking this question is to obtain a thorough discussion of all problems involved in pursuing this type of practice. As is generally known, one of the greatest abuses in the motor-transportation field has been the question of overloading, which seriously affects the general public and has, in the past, resulted in the enactment of a considerable amount of restrictive legislation.

It is my thought that it would be folly for an organization or group of men such as we have here to advocate direct overloading of an individual chassis by the mere

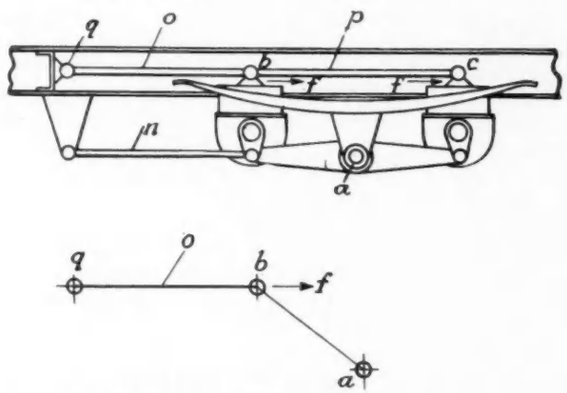


FIG. 6—DIAGRAM SIMILAR TO THAT OF FIG. 5 WITH a AS THE CENTER OF ROTATION

The Direction of Rotation Is as Indicated by Arrows f, the Reaction c Being Transferred to b Through Rod p

application of additional wheels. If we do this we shall mislead many operators who are improperly advised, in some cases, in operating cars. This practice, if pursued, will result in many serious accidents on the highway, and, in view of the serious objection by the general public to large heavy-duty units, the effect of advocating such practice should be carefully weighed.

Satisfactory on Level Road

PIERRE SCHON[10]:—I have seen attachments made to light-duty 1-ton trucks to convert them into 2-ton trucks and, from what the operators tell me, they seem to be doing satisfactory work. In Europe, that type of application has been more popular than in this Country and it has been used for many more years. In a recent transportation-engineering conference attended by 30 representatives of our export company, that subject was thoroughly discussed. Some of our foreign representatives expressed themselves frankly in regard to the conversion of a standard type of motor-truck into a six-wheel truck. The experience in Europe has not always been satisfactory, and I believe that will be true in this Country also. Doubling the capacity places strains on other parts of a chassis which was built originally as a four-wheel chassis. Considerable trouble with the frame, front axle, springs, front-axle knuckles and front wheels has been experienced in Europe after converting light 1½-ton trucks into six-wheel vehicles and using them to carry 4 to 5 tons, as they have been doing.

A number of European manufacturers are supplying those attachments and thousands of Chevrolet and Ford trucks have been converted into six-wheel vehicles in foreign countries. If the operation is confined to level highways having the very best type of pavement, the operation is fairly satisfactory. But the method of balancing the load on a cross-swivel at the rear throws enormous strains on the front axle on rough roads. The difficulties, as described by our foreign engineers, were brought about by the overloads on parts of the chassis unit which were not designed to carry 100 to 200-per cent overload. I believe that these facts apply to trucks of larger capacity also. I think that a truck designed to carry a normal load of 5 tons cannot safely be converted into a 10-ton truck unless certain parts of the chassis are strengthened accordingly.

Popularity and Economy of Six-Wheel Vehicle

There is no question about the future popularity of the six-wheel vehicle. Legislation in various States has restricted the handling of heavy tonnage, but progress of our industries cannot be stayed. Heavy tonnage must be transported and the six-wheel vehicle, whether of the flexible type, such as the tractor semi-trailer, or the rigid six-wheeler, solves many transportation problems. The multi-wheel vehicle complies with many of the State regulations and operators are taking advantage of the additional permissible tonnage.

Economy of operation is apparent, and the cost per ton-mile of handling heavy tonnage is being lowered considerably. This has already been accomplished with the semi-trailer and the four-wheel trailer combinations, and the self-contained rigid six-wheel vehicle will find its field in heavy-duty hauling. I doubt if it will interfere very much with the operation of the flexible type of six-wheel vehicle, namely, the tractor-semi-trailer installation, because each type has its own distinct field.

Apparently a difference of opinion exists regarding the use of a differential between the two drive-axles. Why is a differential considered necessary by one manufacturer and not by others? I know of a number of installations in

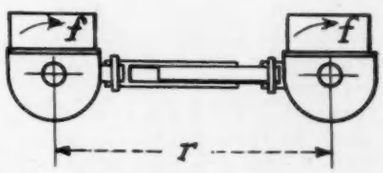


FIG. 7—DIAGRAM SHOWING THE GOODYEAR CONSTRUCTION

The Torque Reaction of the Rear Axle Works Directly Through the Lever-Arm r, Which Is Equal to the Distance Between the Centers of the Axles; the Force Acts To Raise the Front Axle. Similarly, the Reaction of the Front Axle Tends To Depress the Rear Axle

[9] M.S.A.E.—General superintendent of motor-vehicles, Standard Oil Co. of N. J., Newark.

[10] A.S.A.E.—Sales engineer, General Motors Truck Co., Pontiac, Mich.

which six-wheel trucks have been used very satisfactorily for thousands of miles in hauling heavy tonnage without a third differential, in one case for more than 50,000 miles of the most severe service, and without experiencing abnormal axle or tire trouble.

E. W. TEMPLIN[11]:—Well-designed axles are satisfactory without differentials. I believe a conventional type of differential between rear axles of a six-wheel vehicle is out of the question, because when one of the four wheels loses traction the vehicle is stopped, and the main object is to keep going. It is a practical matter. Does it cost more to run without or with a differential? My conclusion is that the construction without a differential is most practical.

C. A. WINSLOW[12]:—On the Pacific Coast I have observed the operation of six-wheel motorcoaches driven by two of the rear wheels. The roads where these vehicles were operating were generally hilly, and the principal trouble was experienced when the roads were wet and slippery. In these instances the traction on the two wheels only was not sufficient to pull the vehicle over the hills.

R. W. KNOWLES[13]:—As Mr. Templin has said, the question is not a matter of individual mechanical practice but of economic results obtained. I think without doubt that we shall see more six-wheel vehicles of the self-contained type in use; also, more semi-trailers. The use of attachments to four-wheel vehicles to convert them into six-wheel vehicles for carrying increased loads has had two incentives; one is to get around the various State limitations with regard to gross weight on the highway; the other is to make possible the carrying of greater load with a single driver and a single investment in truck and body.

Concerning traction, in the oil fields and in places where snow and sleet are likely to be encountered, it is important to have enough space between the driving wheels to prevent "bridging over" in case one axle is on high and the other on low ground; otherwise the vehicle would stall.

Tractor-Trailer Weight-Distribution

W. G. RETZLAFF[14]:—Some years ago we built our semi-trailers with the rear axle near the center of the trailer so that about 33 per cent of the weight was carried on the tractor axle and about 66 per cent on the trailer axle. That worked well in the level States of Ohio, Michigan and Illinois, but in some of the eastern territory we found that, especially after a light snowfall which turned to rain, the tractors often did not have sufficient traction to climb the hills; if they did surmount the hills and their brakes were applied to come down, they would slide. Therefore we redesigned the weight distribution on the axles to be about 45 per cent of the pay-load on driving or truck axle and 55 per cent on the trailer axle. The extra weight of the rear-end of the truck is an additional 5 per cent; hence, about 50 per cent of the gross load is carried on the tractor rear-axle and 50 per cent on the trailer axle. We have not had a single complaint of slippage and danger in applying the brakes for coming down the hills in either the eastern or any other mountainous territories.

Long-Distance Freight Haulage

In handling special commodities such as gasoline and milk, the semi-trailer is a very efficient unit, but we now find that some of the long-distance-freight haulers are competing successfully with the railroads and hauling first, second, third and fourth-class general freight at railroad rates up to distances not exceeding 300 miles, using the tractor-trailer method. Some operators tried several years ago to use the heavy 5-ton truck to haul general freight at railroad rates over the highways, but the weight restriction was such as to limit them to about 6 or 7-ton pay-loads and they could not make it pay. The next idea was to use a 3½-ton high-speed tractor and haul a 10-ton pay-load at an average speed of about 35 m.p.h. So long as they hauled a 10-ton pay-load both ways they could meet the railroad rates and make the operation pay; but in most cases they hauled a pay-load one way and could secure only a light return load, so the average pay-load was about 7½ tons and they did not find that very profitable.

The next step was to use tractors having a higher gear-ratio in the rear axle to pull both a semi and a four-wheel trailer and double the pay-load. The tendency now is, especially on level roads, to use a heavy-duty 3½-ton tractor having a low gear-ratio, which probably cannot exceed a speed of 25 m.p.h. The semi-trailer is easy to operate into and out of difficult places, but the four-wheel trailer is hard to handle.

Pick-up and delivery costs also should be considered in long-distance freight hauling. Some haulers say that these costs exceed their long-distance freight-haulage costs. In such cases the four-wheel trailer is objectionable because it is hard to handle when delivering freight. Another plan is to haul two semi-trailers in a train, the rear trailer being converted into a four-wheel unit by means of a dolly. The best plan is to operate the tractor-trailer equipment on the long-distance runs at night and have it arrive early in the morning. Another driver then takes the unit and the freight is shifted or delivered. A new load is picked up in the afternoon, and the train makes a return trip that night. With the four-wheel trailer it is necessary to shift the freight to a lighter unit at the depot and then make the deliveries, because all of the consignees have not the facilities for picking up their freight. With the semi-trailer, however, this freight-shifting to smaller delivery units is not necessary. Therefore, there might be some advantage in using the six-wheel truck as a tractor and applying four wheels under the semi-trailer. We would then have a unit which would be easy to handle, and would be able to double the load. No doubt the ideal unit for long-distance freight operation will at some future time be the six-wheel tractor with a four-wheel semi-trailer.

Possibilities of Dual Front-Wheel-Tire Usage

R. E. PLIMPTON[15]:—The suggestion has been made to put dual tires on the front wheels. How far has that been developed as practical construction?

MR. WOLF:—The problem is purely one of steering ease. A number of such applications have been made, as shown in Fig. 14 of my paper. The steering-knuckle-pivot axis usually is located so as to give center-point

[11] M.S.A.E.—Formerly, motor-truck engineer, Goodyear Tire & Rubber Co., Akron, Ohio.
[12] M.S.A.E.—Development engineer, Hercules Motors Corp., Canton, Ohio.
[13] M.S.A.E.—Transportation engineer, The White Co., Cleveland.
[14] M.S.A.E.—Transportation engineer, Fruehauf Trailer Co., Detroit.
[15] M.S.A.E.—Associate editor, *Bus Transportation*, Chicago.

steering with reference to the inner tire. The center distance between the tires is then the lever-arm through which the effort must be exerted to swing the outer tire and wheel, other than the friction caused by the road contact of the inner tire.

Should there be sufficient demand for dual tires, one truck manufacturer is willing to locate the steering-knuckle-pivot axis midway between the centers of tire contact with the road to minimize the steering effort. Of course, a slight offset from the actual mid-position is desirable to assure the taking up of all slack and the prevention of the central "wandering" effect that would otherwise occur.

CHAIRMAN WINCHESTER:—Is anything being done on the conventional design of the rear-wheel drive along similar lines?

MR. WOLF:—Not so far as I know.

Operating Experiences Related

F. C. MCMANUS[16]:—Our first six-wheel vehicle had a conventional differential, and the question arose as to what could be done if one wheel should spin. The next step was to put a solid shaft between the axles, similar to the Timken or the Safe-Way construction. It has been brought to our attention that, where the differential is omitted, it is possible to break a jackshaft or axle shaft in one of the axles and continue operation without knowing about it; that is, the drive will be continually on the other axle. I understand that in some cases this has gone along for two months. In one truck that has no differential between the jackshafts a key in the rear universal-joint sheared off and the front jackshaft did all the work until other trouble occurred and the broken key was discovered.

Slackness of the chains in the chain-drive construction provides a differential action which most persons overlook; it is comparable with the spring-loaded torque-arms in the construction shown by Mr. Buckendale. I believe that the drive chains provide as much differential action as do the spring-loaded torque-arms.

Following the differential of the conventional type which we used at first, a friction-type differential something like that described in Mr. Wolf's paper was brought out in which the total torque through the third differential can be regulated to throw perhaps 55 to 75 per cent of the normal torque on the one axle which is driving, provided that the other axle builds up sufficient resistance to make the differential operate.

I should like to emphasize that at only one time is the drive on all four wheels needed; that is when the vehicle is stuck in a hole and the driver uses first or second speed, never third or fourth speed. For such cases it might be possible to incorporate some device similar to the Timken power-lock that automatically would become inoperative after the driver changed out of second speed. That might solve the problem of stalling all four wheels in a bad spot and still provide the actual differentiation between the jackshafts to take care of the difference in diameter of tires and also the surmounting of ordinary obstructions. As reported by Ethelbert Favary recently, the difference between the track radii of the rear-end front and rear tires of the same size is only ¾ in., and this ought to be taken up normally by pneumatic tires. The main thing we must watch for is the difference in tire sizes between the two driving-axles of the rear-wheel unit. I think none of the tire companies make tires of the same capacity that have exactly the same rolling radius; in some cases the difference doubtless amounts to ½ to ¾ in. on some of the large sizes. In case the tires on the rear-wheel unit have different rolling radii, the larger tires will soon wear down to the size of the smallest one, which no one can afford.

L. R. BUCKENDALE[17]:—The question of a differential between the two rear driving-axles is a subject that is debatable at present. Opinions are both for and against it. The final answer undoubtedly must develop from experience and trial. The question is whether that additional differential complication will in turn be repaid by the better results, such as in increased tire life and better distribution of stresses.

Over-All Vehicle-Width Handicaps Design

CHAIRMAN WINCHESTER:—The question of over-all vehicle-width is very serious, particularly on the heavy units. In the majority of cases the State laws now set a limit of 96 in. For a certain type of vehicle to run on pneumatic tires and carry a 10-ton load, the narrowest vehicle we recently could obtain on the market was 98 in. wide, and the widest one having the necessary tire capacity was 115 in. wide; finally, we had to use solid tires to keep the over-all width within 96 in. to conform with the laws in certain States.

Within the last few years an actual reduction has been made in the load-carrying-capacity tables of tire companies. On the other hand, the companies manufacture so-called super tires which are supposed to give much greater mileage than that of the standard tires sold. What relation does the super tire have to the particular problem of giving an increased tire-capacity rating? Mr. Sprowls makes the point that balloon tires might be advisable in this type of operation, but if balloon tires are considered the over-all width would be even greater.

We have operated both types of drive. Recently we tested a light unit on which we wanted to use single instead of dual tires to secure a narrow over-all width. We installed a six-wheel attachment, thus getting the benefit of dual tires but using two axles. The width was very narrow, but the problem of traction was serious because the power was applied only on the forward axle. The point raised by Mr. Shields about driving from the rear instead of from the front rear-axle seems original and one that might be tried out profitably.

MR. SHIELDS:—Does Mr. McManus find it more common to experience trouble on six-wheel jobs not equipped with differentials between the rear axles?

MR. MCMANUS:—Yes; it is possible that the trouble may be with an axle shaft or some broken part inside. There are times when the full load is thrown on one axle, and I am not certain that anyone will build the parts strong enough to take the full load of the engine on one axle.

MR. SHIELDS:—In one case the differential heated to such a degree that it seemed to be burning the oil. The trouble was caused by one of the tires, which had been run while quite flat. When the tire was inflated to normal pressure, the heating of the differential ceased, which emphasizes the point that the drag on the tires, if they have different rolling-radii while operating,

[16] Jun. S.A.E.—Engineering department, International Motor Co., New York City.

[17] M.S.A.E.—Executive engineer, Timken-Detroit Axle Co., Detroit.

represents a tremendous consumption of power, and that this wastage and destruction must be eliminated.

When balloon tires are installed on six-wheel vehicles that are not equipped with differential units—and I think balloon tires will be standard in the near future—more trouble can be expected than when using either high-pressure or solid tires, because there is more likelihood of a greater variation in loaded radius and in air pressure, and the greater contact with the ground will put a greater load on the engine because the tires will not slip so readily.

Maximum Dimensions for Nominal Tire-Sizes

MR. SPROWLS:—Regarding the statement that the sizes of tires of given capacity made by different manufacturers vary, the Tire and Rim Association recently has standardized on sizes in that it has decided on the maximum dimensions for each nominal size. I think most of the manufacturers either have taken or are taking steps to make their tires conform to this standard; so I believe no future trouble will be caused by difference in sizes.

In considering the possibilities of having only one of the two rear axles of a six-wheel vehicle used as a driving axle, we have had some experience that may prove illuminating. We discovered fast tread-wear on the center tires of a six-wheel truck which had a differential on both the rear and the central axle but no third differential between the two axles. We could not understand the reason for this tread-wear until we finally jacked up both the rear and the central axles and ran the engine. We then discovered that the central axle was turning and the rear axle was not. Upon further examination we found that the driveshaft between the central and the rear axle was broken. We therefore attributed the fast tread-wear to the fact that all of the driving had been done by the tires on the central-axle wheels. This fact should be considered in centralizing driving on one axle.

In regard to obtaining super tires of larger carrying-capacity, I am doubtful that we shall be able to increase the carrying capacity, although we may be able to increase the mileage. Most of the super tires made today have much thicker treads, but they do not have greater carrying-capacity, which is dependent upon the amount of air in the tire.

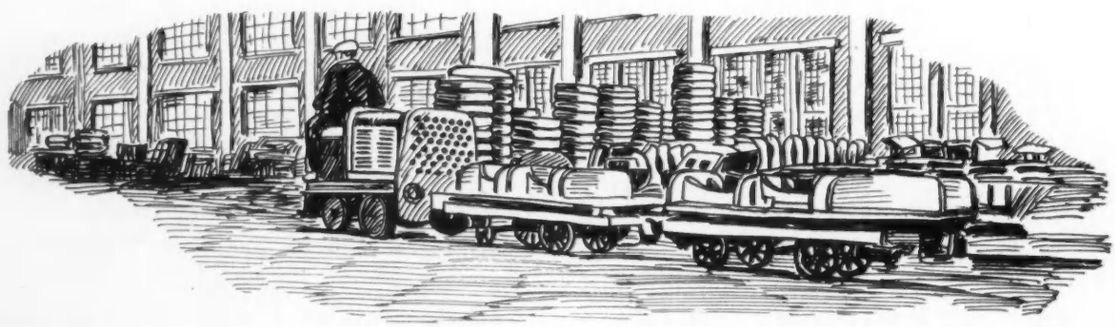
MR. KNOWLES:—Regarding the impact and the loading of the four-wheel as compared with the two-wheel drive, the pressures shown were very much lower on the former than on the truck of conventional type carrying a corresponding tonnage. How does the tire wear compare with that of a similar tire on the rear of a semi-trailer? There might be a marked difference, because the truck axles are driving while the trailer is not.

MR. RETZLAFF:—Even though the tractor tires do the driving, the trailer tires will wear out first because the operator invariably will load his heavier freight on the rear end. Theoretically, the weight distribution is 50 and 50 per cent but, if weighed physically, it is more likely to be 35 and 65 per cent. This overloading of the trailer tires results in their shortened life.

Danger in Converting Vehicles

CHAIRMAN WINCHESTER:—Greater use probably has been made of the six-wheel truck and six-wheel attachment on the Pacific Coast than in any other section of this Country. In the last legislative meeting in California, a distinct tendency existed to lower the gross-load limits allowed for trucks in that State. There seemed to be a conflict between two outstanding groups; one which manufactures the six-wheel attachment and one that manufactures so-called self-contained six-wheel trucks. It is lamentable that the two groups cannot agree on what is best. I can understand the individual viewpoint of each group, because each has its living to make, but some of the propaganda circulated is to the detriment of the industry as a whole. I fear that these 3-ton and 5-ton trucks which are converted into six-wheel units will in many cases have no additional braking power and no increase in strength of the frame. This will result in accidents and a general cry against the use of heavy-duty units. That sort of thing must be studied and avoided as far as possible.

We are discussing six-wheel trucks on the basis that they can be operated everywhere to advantage, but that is not true. Some States are now in a receptive mood toward this type of vehicle; however, they will not receive it with favor unless it is advanced on logical and fair grounds. As an organization and as a representative group of citizens who understand the engineering features of these problems, we should take great interest in the subject from a civic viewpoint. We not only manufacture the trucks and operate them, but we are also responsible to a large degree for the roads on which they operate. If we fail to design the vehicle along logical lines, we shall ultimately place a burden upon the entire industry that will be difficult to assume. Exceedingly heavy loads on improperly designed vehicles will result in breakdowns of the highways that will react against the entire industry.



Friction-Coefficient Research

By Louis Illmer[1]

1930 Annual Meeting Paper · *Illustrated with* Charts

This paper is a comprehensive survey of representative experimental research directed toward perfect-oil-film lubrication. With this as a basis, an endeavor is made to correlate empirically, by means of analytical research, the experimental results obtained by different investigators and to establish certain underlying principles common to all such tests in which the frictional resistance depends primarily upon fluid shear.

The conclusions arrived at are predicated upon a rather wide range of reported determinations, including most of the relevant tests that have found their way into friction literature during the last 50 years. In piecing together this accumulated experience, along the lines suggested some years ago by Prof. M. D. Hersey[2], a reasonably reliable but relatively simple method for predetermining certain kinds of friction losses has been deduced.

A JOURNAL is subjected to little or no metallic wear when riding upon a perfect or unbroken oil film, and the relatively low friction drag becomes substantially independent of the rubbing materials if they are given a sufficiently smooth finish. Under such favorable conditions, the resulting friction coefficient affords a suitable standard of reference with which to compare the behavior of the more involved or mixed type of friction.

Friction coefficients pertaining to perfect-film lubrication depend upon a number of controlling factors, many of which have been either ignored or not adequately taken into account in the majority of reported friction determinations. An object of the inquiry herein reported is to find a functional relationship between these controlling factors that shall be directly proportional to the friction coefficient sought. The present study undertakes by empirical algebraic treatment to approach the indicated analysis along somewhat original lines without, however, making use of any particular theory of lubrication. Instead, the given coordinated results depend wholly upon the cited friction-test data. Only that portion of the experimental work of others which furnishes sufficient detail or permits the data to be reasonably interpolated has been utilized.

For each such test series, a separate equation has been deduced that is intended to be fairly representative of the test in question. This method, applied to a rather wide range of determinations, shows certain inconsistencies, and it has been necessary to seek further corrective factors that might reconcile and bring these seeming discrepancies into harmonious relation. In this connection, it is to be remembered that the conventional friction coefficient is not a physical constant in a rational sense, but merely represents a convenient measure of bearing efficiency.

A list of the rotating-journal tests that have been considered is presented in Table 1, which gives a digest of the more important factors that could be gathered from the accompanying reports. Each such test is assigned a distinctive identification number in the first column, and an equation is given in the second column for the corresponding friction coefficient, designated as f.

Analyzed Journal Tests

The succeeding columns indicate the grade of oil used, the rate of feed, and an approximate equation for its absolute viscosity μ. The journal fit is reported in one column, but this important factor has been found missing in many instances. Other columns specify the journal size, the character of the rubbing materials, and the velocity and pressure range under which the respective tests were run. The factors which fix the coefficient of friction f, center around viscosity, μ; velocity V; and the specific pressure intensity P; all of which units are more fully defined in the accompanying list of symbols.

In Fig. 1 have been plotted, for an assumed velocity of 100 ft. per min. and an assumed average absolute-viscosity value of 0.5, those equations for f in Table 1 whose actual test values fall within the stipulated plot-range. A composite or average-value curve was then drawn in heavy line among these plotted curves and designated as f_j.

Fig. 2 was arrived at in a similar way, except that the velocity was assumed to be 400 ft. per min. Fig. 3 is based upon a velocity of 1000 ft. per min., which is the highest speed for which a sufficient number of comparative equations could be found. Why certain of these test lines lie considerably above or below the respective composite values will be explained later.

Basic Equation for Coefficients

For purposes of further discussion, the general form of equation suitable for present purposes may be written, where f is greater than f_{min},

$$f = C_1 C_2 \cdot \frac{1 \mu^r V^m}{(P/S)^n} \qquad \text{(D1)}$$

[1] M.S.A.E.—Development engineer and patent attorney, The Brewer-Titchener Corp., Cortland, N. Y. The author wishes to express appreciation of the assistance rendered by his wife, Winifred R. Illmer, in the preparation of this paper.

[2] See Transactions of the American Society of Mechanical Engineers, vol. 37, 1915, p. 197.

SYMBOLS USED IN THIS PAPER

C_1 = type constant as deduced from experimental data, about 1/7 to 1/8 in the case of journals that are borne upon an unruptured oil-film

C_2 = circumstance constant, the numerical value of which is never less than unity and the purpose of which is discussed in connection with Table 4

C_3 = viscosity constant

D = nominal diameter of journal, inches

E_f = fluid-shear efficiency for any given pressure intensity when operating under different values of exponent n, when $P < P_c$

F = friction mean effective pressure as transferred to the effective working stroke; it measures internal engine-losses such as piston drag or crankshaft or valve-gear friction per square inches of piston area. Cylinder pumping-losses are not included.

f = nominal coefficient of friction for substantially perfect oil-film lubrication, particularly as applied to a rotating journal and taken with respect to the projected pressure intensity P/S, defined below

f_1 = journal-friction coefficient corresponding to perfect film formation when the factors C_2 and n are unity

f_c = journal-friction coefficients for substantially unbroken film formation for values of $n < n_0$ or $K < K_0$

f_{exc} = coefficient for intermediate friction as obtained under conditions of partial-film lubrication

f_j = composite or average value of the friction coefficient f as derived from the test data plotted in Figs. 1 to 3

f_k = coefficient of friction based upon a test in which the fit factor K is made closer than K_0

f_{min} = minimum value for the coefficient f as reached when the journal is subjected to the pressure P_c

f_p = preliminary value for the friction coefficient usually attained in loosely fitted journals sustained upon a substantially perfect oil-film

f_r = coefficient of friction pertaining to railway-car-journal practice

f_s = equivalent friction coefficient derived from the roller-bearing tests

H = specific radiating capacity per square inch of projected journal area, foot-pounds per minute

K = constant depending upon the kind or class of fit used

K_0 = critical value of the fit constant K above which the character of oil-film lubrication is modified in that the use of a closer journal-fit no longer correspondingly increases the contact factor S

L = bearing length, inches

l = mean center lift of the journal as raised by the interposed oil-film, inches

l_1 = vertical lift component of the Goodman journal center as raised by the interposed oil-film, inches

max = subscript to designate maximum value of the symbol to which it is attached

min = subscript to designate minimum value of the symbol to which it is attached

N = journal speed, revolutions per minute

n = pressure exponent

n_0 = bearing-characteristic or limiting value of the pressure exponent n that can be reached under any given fit conditions

m = velocity exponent

P = mean-nominal or virtual unit-pressure carried by an oil-borne bearing, in pounds per square inch of projected journal area as measured over the entire journal diameter

P_1 = equivalent pressure for roller-bearings in terms of projected journal area, pounds per square inch

P_c = critical pressure at which the friction coefficient assumes a minimum value, in pounds per square inch of projected journal area

P/S = mean pressure intensity actually prevailing in the bearing, pounds per square inch of projected journal area

r = viscosity exponent

S = surface-contact factor or fraction of the full projected journal area that is effectively utilized

S_0 = maximum value of the surface-contact factor attainable under given operating conditions

T = bearing running temperature, degrees fahrenheit

T_0 = virtual temperature factor or characteristic of any given oil, degrees fahrenheit

t_h = temperature head of the rubbing surface with respect to the surrounding cooling medium, degrees fahrenheit

V = mean rubbing velocity of the rotating journal, feet per minute

V_c = critical journal-surface velocity at which the resulting film lifting-pressure becomes equal to P_c, feet per minute

α = semi-arc length of bearing bore that is brought into effective journal contact, degrees

β = marginal oil-film thickness at the extremity of the arc α

δ = total fit allowance or difference in diameter between the journal and the bearing bore, inches

μ = absolute viscosity of the lubricant as determined at atmospheric pressure for any given bearing temperature, in centimeter-gram-second units or poises; equals kinematic viscosity $\times$ specific gravity

$\approx$ = approximately equal to

This equation applies only when the defined pressure P is kept lower than the critical pressure P_c which the oil film is capable of sustaining without breakdown, and may be regarded as a standard of reference for the minimum expected coefficient under given operating conditions. A distinctive feature of equation (D1) is

that the virtual-pressure factor P is found below the line of the fraction, as its denominator. Hence, when working with true-film lubrication, the coefficient of friction will diminish with increased unit pressure until the critical pressure is reached; that is, until the friction coefficient reaches the minimum. The exponent n fixes the accompanying rate of friction reduction; it may take on a value considerably below unity and afford a relatively low resulting friction coefficient that is still representative of true film formation.

Negative Exponent Shows Imperfect Lubrication

Perfect-film-borne journal friction is characterized by a positive value of the pressure exponent n. It will later be shown that this exponent usually assumes a low value when the shaft fit is loose and the contact factor is given a correspondingly low value. This exponent may also assume a negative value, whereupon the friction coefficient becomes directly instead of inversely proportional to P. This in turn denotes imperfect friction conditions, thus indicating that the oil film has partly broken down and that we then have to do with intermediate or partial-film lubrication. The latter phase has been more specifically treated in an earlier paper by me, on High-Pressure Bearing Research[3], which also reveals that friction of the f_{exc} character is lowered with increased velocity, instead of being augmented therewith in the way represented by equation (D1).

The numerical value for the critical pressure P_c is found to depend upon the journal rubbing velocity, the oil viscosity, and the character of the journal fit. Equation (D1) holds good only when the V and μ factors are kept sufficiently high to allow establishing an oil film that will floatingly uphold the unit-pressure intensity P/S. A rubbing velocity higher than 100 ft. per min. usually suffices to exert a lifting pressure of several hundred pounds per square inch of projected journal area.

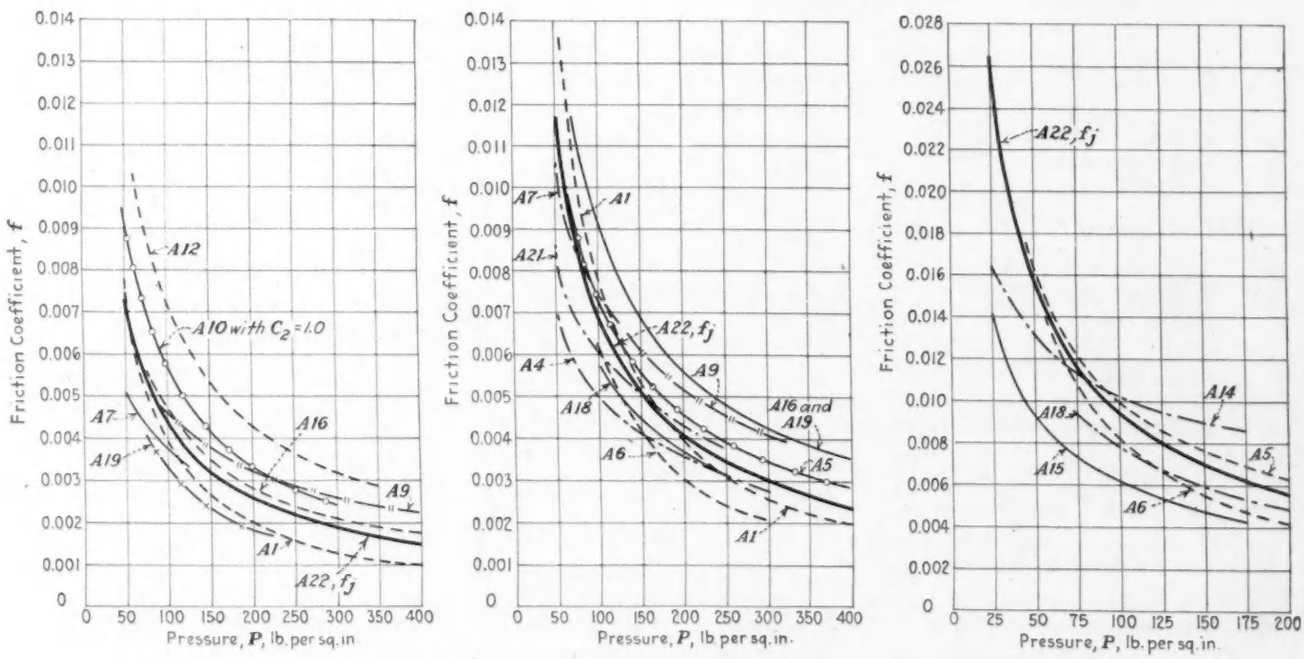
If the surface speed should lie below the critical velocity V_c and no longer provide for the required lifting effect, the resulting friction will be of the imperfect or intermediate type, for which the coefficient is likely to be appreciably higher than when the journal is carried upon a perfect oil-film. It will therefore be understood that the lowermost range of equation (D1) becomes invalid for any coefficient that may fall below f_{min}. Under the conditions stipulated in Figs. 1 and 3, this critical value may be expected to lie between 0.002 at lower velocity and 0.004 at higher velocity.

Behavior During Starting

When a journal starts to move from rest, the accompanying f_{exc} friction coefficient at first drops at a rather rapid rate with increased speed until f_{min} is reached at velocity V_c, but thereafter the value of f follows a new law and increases with augmented velocity in accordance with equation (D1). Conversely, at any fixed journal speed lying above the critical velocity V_c, the friction coefficient f will be lowered with increased pressure until the f_{min} friction value is approached; and, should the pressure be carried beyond its critical P_c value, the friction coefficient will again enter the intermediate or ruptured-oil-film region. These aspects will be more fully treated in a subsequent paper, which will deal more specifically with the load-carrying capacity of oil films and minimum friction coefficients.

Referring further to Figs. 1 to 3, the various plotted curves show that all of the stipulated exponential values

[3] See Transactions of the American Society of Mechanical Engineers, vol. 46, 1924, p. 833.



FRICTION CURVES FOR VISCOSITY $\mu = 1/2$ AND VARIOUS RUNNING SPEEDS

Curves Are Numbered According to the Test Numbers in Table 1. The Heavy-Line Curve Is from the Composite Equation

Fig. 1—100 Ft. per Min. Fig. 2—400 Ft. per Min. Fig. 3—1000 Ft. per Min.

TABLE 1—BEARING DATA PERTAINING TO

Reference Test No.	Experimenter [a]	Equation for Average Friction-Coefficient	Lubricant Data: Kind Used	Lubricant Data: Rate of Feed	Lubricant Data: Viscosity Equation	Journal Clearance, δ, In.
A1	Beauchamp Tower	$f = 0.055 \frac{\sqrt{(\mu V)}}{P}$	Various oils	Bath		0.0035 [b] (close fit)
A2	Beauchamp Tower	$f = 0.06 \frac{\mu^{1/2} V^{1/3}}{P^{2/3}}$	Rape oil	Siphon lubricator (slow drop-feed)	$\mu = \frac{21}{T - 50}$ [b]	Probably same as in A1.
A3	John Goodman	$f = 0.046 \frac{V^{1/2} \mu^{1/3}}{P}$		Bath	$\mu = \frac{18}{T - 50}$ [b]	
A4	R. Stribeck	$f = 0.004 \mu \sqrt[2/3]{\left(\frac{V}{P}\right)}$	Motor gas-oil	Ring oiler	$\mu = \frac{22}{T - 70}$	0.0072 [b] (normal fit)
A5	R. Stribeck	$f = 0.034 \frac{\mu^{2/3} V^{1/3}}{P^{2/3}}$	Motor gas-oil	Ring oiler	$\mu = \frac{22}{T - 70}$	0.0029 [b] (close fit)
A6	O. Lasche	$f = 0.13 \frac{\mu^{1/2} V^{1/3}}{P}$	Imperial No. 0	0.21 gal. per min. ring oiler	$\mu = \frac{10}{T - 60}$ [b]	0.010 [b] (fast-running fit)
A7	C. C. Thomas E. R Maurer and L. E. A. Kelso	$f = 0.0053 \sqrt{\left(\frac{\mu V}{P}\right)}$	Atlantic red engine-oil	Ring oiler	$\mu = \frac{20}{T - 60}$ [b]	
A8	W. F. Parish	$f = 0.08 \frac{\sqrt{\mu}}{\sqrt[3]{P}}$	Texas Cetus	Various	$\mu = \frac{20}{T - 50}$ [b]	
A9	James Hopps	$f = 0.012 \frac{\sqrt[3]{(\mu V)}}{\sqrt{P}}$	Various oils	Bath		Close fit
A10	H. Heinmann	$f = 0.055 \frac{\sqrt{(\mu V)}}{P^{3/4}}$	Bearing No. 1	Free, cup feed	$\mu = \frac{20}{T - 60}$ [b]	0.0024
A11	A. Martens	$f = 0.0045 \frac{\sqrt{(\mu V)}}{P^{1/3}}$	Various oils		$\mu = \frac{20}{T - 50 \text{ to } 100}$	
A12	Albert Kingsbury	$f = 0.012 \left(\frac{\mu V}{P}\right)^{3/4}$	Lard and olive		$\mu = \frac{18.5}{T - 50}$ [b]	
A13	Albert Kingsbury	$f = 0.016 \frac{\mu^{1/3} \sqrt{V}}{P}$	Heavy machine	Flooded	$\mu = \frac{20}{T - 60}$ [b]	0.030 for large and 0.0075 for small
A14	D. S. Woods and D. P. Carter	$f = 0.006 \sqrt[3]{\left(\frac{\mu V}{P}\right)}$	Light mineral	Free, cup feed	$\mu = \frac{17}{T - 55}$ [b]	
A15	G. Stoney R. O. Boswell and J. Massey	$f = 0.005 \frac{\mu^{7/16} V^{1/2}}{P^{5/8}}$	Turbine gear	Submerged, bath	$\mu = \frac{14}{T - 65}$	Close fit
A16	L. Ubbelohde	$f = 0.012 \frac{\mu^{1/3} V^{1/2}}{P^{2/3}}$	Various oils		Values given	
A17	H. Nielsen and Stanley Baker	$f = 0.075 \sqrt{\left(\frac{\mu}{P}\right)}$	Coal-distillate oil	Free, pipe feed	$\mu = \frac{12}{T - 95}$	About 0.009
A18	H. A. S. Howarth and Nelson Ogden	$f = 0.024 \frac{\sqrt{(\mu V^{2/3})}}{P^{2/3}}$	Machine oil		$\mu = \frac{12}{T - 60}$ [b]	Six tilting thrust-bloeks
A19	M. D. Hersey	$f = 0.0055 \left(\frac{\mu V}{P}\right)^{3/4}$	Various oils	10–60 drops per min.	Values given	0.004
A20	T. E. Stanton	$f = \frac{1}{710} \cdot \frac{\sqrt{(\mu V^{2/3})}}{P^{1/4}}$	Various oils	Flooded	Values given	0.02 and 0.06
A21	John Goodman	$f = \frac{1}{550} \frac{\mu^{3/8} V^{5/8}}{\sqrt{P}}$	Castrol A	Bath	$\mu = \frac{40}{T - 60}$	0.035, as run
A22	Composite	$f_j = \frac{\sqrt{(\mu V^{2/3})}}{24 P^{3/4}}$	Any kind	Flooded	No fixed limit	Normal horizontal-journal fit

[a] References to publication or source of information are listed on p. 72, arranged according to designations in the first column.
[b] These values have been computed or interpolated from given test data, as they are not recorded.

FILM LUBRICATION OF JOURNALS WHEN $P < P_c$

Journal Size, In.		Materials	Velocity Range Ft. per Min.		Pressure Range Lb. per Sq. In.		L/D	Temperature Range Deg. Fahr.		Absolute Viscosity Range		Remarks	Reference Test No.
Diam.	Length		$V_{min.}$	$V_{max.}$	$P_{min.}$	$P_{max.}$		$T_{min.}$	$T_{max.}$	$\mu_{min.}$	$\mu_{max.}$		
4	6	Steel on bronze	105	470	100	625	1.5	90	90	0.22[b]	1.0[b]	Brass on top of journal. Equation based on all oils tested except rape	A1
4	6	Steel on bronze	157	366	123	317	1.5	80	90	0.52[b]	0.70[b]	Chord of contact arc = 0.81 D	A2
2	4	Steel on bronze	8	123	37	150	2.0	68	104	0.33	1.00	Pressure changed by varying block chord from 1/2 to 2 in.	A3
2 3/4	9 1/8	Steel on cast iron	550	800	6	215	3.3	75	140	0.32	2.0	Loosely fitted cast-iron shell	A4
2 3/4	5 3/8	Steel on babbitt	272	800	34	320	1.96	68	117	0.56	2.5	Closely fitted babbitt shell	A5
10 1/4	4 3/8	Steel on babbitt	200	3000	15	215	0.42	85	210	0.07[b]	0.40[b]	Cooled oil fed into top of bearing cap	A6
2 7/16	9 5/8	Cold-rolled steel on babbitt	96	287	33	100	3.9	80	110	0.40[b]	1.00[b]	Tests made upon conventional line-shaft bearings	A7
3[b]	8	Steel on babbitt	156[b]	156[b]	14	181	1.9	104	155	0.19[b]	0.37[b]	Oil feed varied, given f for 3/4 pint per hr. or more; test run upon Thurston machine	A8
1 (vertical)	2	Various	100	400	100	800	2.0	95	95	0.2[b]	2.6[b]	Given f for phosphor bronze on case-hardened journal; chord of contact arc = 0.81 D	A9
1 3/16	2	Steel on bronze	40	670	43	160	1.7	68	122	0.32[b]	2.5[b]	Tests made with larger bore clearances, not consistent	A10
3 7/8	2 3/4	Steel on bronze	100	400	140	570	0.7	70	105	0.30	2.0	Absolute viscosity values given	A11
1 3/8 (vertical)	2	Steel on bronze	43	101	40	340	1.5	90	170	0.15[b]	0.46[b]	f values exceptionally low at higher pressures	A12
9 15			520 (averages)	1390	82	144	3.0 (average)	100 (averages)	129	0.29[b]	0.50[b]	f remains constant above critical velocity of 1000 ft. per min.	A13
9	7[b]	Steel on bronze	470	2240	50	150	0.8	70	140	0.25[b]	1.20[b]	f remains constant above critical velocity of 1200 ft. per min.	A14
2 1/2	4	Steel on babbitt	1700	3800	40	120	1.6	100	140	0.16	0.42	Chord of contact arc = 0.7 D	A15
			118	452	99	469		72	100	0.45	3.43	Used Marten's machine having separate blocks set 120 deg. apart.	A16
2	4	Steel on bronze	680	680	400	800	2.0	104	194	0.10	1.30	Journal clearance abnormally large	A17
11 1/2 (mean)		Iron on babbitt	180	780	100	300		77	131	0.70	1.70	Test made upon Kingsbury segmental thrust-bearing	A18
1	3	Steel on bronze	145	625	40	255	3.0			0.38	4.40	Thesis test reported by R. E. Wilson and D. P. Barnard, 4th	A19
1	2 1/2	Steel on bronze	262	262	277	277	2.5	90	120	0.25	2.09	Tests run on small circumferential-contact area	A20
6	3	Steel on bronze	157	700	28	222	0.5	60	160	0.40	5.4	Tests run with full 180 deg. shell-area	A21
			40	1500	$P < P_c$		1.5+	68	300	0.10	4.00	Composite value	A22

may vary widely without materially altering the resulting friction coefficients, because such exponents merely measure the accompanying rate of change. In some instances, these exponents might be changed considerably and still conform fairly well with the experimental findings.

In locating the heavily drawn composite line, due weight was given to the different tests in accordance with their apparent trustworthiness. The deviations from normal are not unreasonable, considering that the higher f values are in part due to C_2 values greater than unity, while certain of the low values may be accounted for by correspondingly low contact factors S.

The character of fit between the rubbing surfaces has a decided influence upon the friction coefficient, as will be shown presently, but this factor is seldom measured and reported. The Stribeck tests, upon which equations (A4) and (A5) are based, have proved to be among the most satisfactory for present purposes.

It is to be emphasized that the nominal unit pressure P is here taken over the entire projected journal area. When this is divided by the contact factor S, the resulting pressure becomes a measure of the actual specific pressure intensity that prevails in the bearing. Not all of the projected area is likely to be utilized when the journal fit is made relatively loose; nor is the film formation quite so perfect as is maintained in the case of the more closely fitted journal.

It will be noted further that the viscosity exponent, as evaluated by the composite equation (A22), has been fixed at $r = \frac{1}{2}$; that is, the analyzed tests show the friction coefficient to increase with the square root of the absolute viscosity of the lubricant rather than directly with it.

The present findings fix the velocity exponent m at 1/3. This investigation rather conclusively proves that too much weight is usually given to the velocity factor. In moderate-speed practice, for instance, the coefficient of bearing friction may be assumed to remain substantially constant for the accompanying velocity range, since the relatively small decrease in friction that might be expected to accrue at slower speeds apparently is lost under ordinary conditions of commercial operation. The given m value is especially applicable when the value of n is less than 1, as it generally is in practice.

TABLE 2—VISCOSITY FACTORS OF COMMON OILS

Grade	Constants C_3	Constants T_0 (Deg. Fahr.)	Kind of Oil	Viscosity $\mu =$
A	1–5	10–40	Water and like thin fluids	
			Velocite oil	$5/(T-40)$
B	5–12	40	Spindle oils	
			Sperm	$11/(T-40)$
			Light paraffin	$13/(T-35)$
C	10–20	50	Light machine-oils	
			Olive	$18/(T-50)$
			Lard	$19/(T-50)$
			Rape	$20/(T-50)$
D	15–25	60	Medium machine-oils	
			Engine oil	$20/(T-60)$
E	20–35	70	Heavy machine-oils	
			So-called motor gas-oil	$22/(T-70)$
			Russian mineral	$28/(T-70)$
			Castor-oil	$65/(T-70)$
F	42	85	Heavy lubricant	
G	50	100	Steam-cylinder oils	

The experimental data herein considered, for the most part, show the coefficient f to be directly proportional to some root of the surface velocity V, rather than to the angular velocity or journal speed N. Furthermore, the preferred V measure has been adopted quite generally in experimental reports and is also better suited to bearing-design purposes.

Before discussing Table 1 in further detail, it is

LIST OF REFERENCES FOR EXPERIMENTS CITED IN TABLE 1, ARRANGED ACCORDING TO REFERENCE TEST NUMBERS

*A*1—First report, Proceedings of the Institute of Mechanical Engineers, November, 1883, p. 632; second report, Proceedings of the Institute of Mechanical Engineers, January, 1885, p. 58.

*A*2—Same as *A*1.

*A*3—Friction and Its Reduction; by G. U. Wheeler, 1903 edition, pp. 43-46; Macmillan Co.

*A*4—*Zeitschrift des Vereines Deutscher Ingenieure*, Sept. 6, 1902, p. 1341.

*A*5—*Zeitschrift des Vereines Deutscher Ingenieure*, Sept. 20, 1902, p. 1432.

*A*6—*Zeitschrift des Vereines Deutscher Ingenieure*, Dec. 13, 1902, p. 1881.

*A*7—Transactions of the American Society of Mechanical Engineers, vol. 35, 1913, p. 593.

*A*8—Transactions of the National Association of Cotton Manufacturers, Oct. 2, 1913, p. 255.

*A*9—*Engineering* (London), Nov. 2, 1906, p. 594.

*A*10—*Zeitschrift des Vereines Deutscher Ingenieure*, July 15, 1905, p. 1161.

*A*11—Mitteilungen aus den Königlichen Technischen Versuchsanstalten zu Berlin, 1889.

*A*12—A Treatise on Friction and Lost Work in Machinery and Millwork; by Robert H. Thurston, 1903 edition, p. 398; John Wiley & Sons, New York.

*A*13—Transactions of the American Society of Mechanical Engineers, vol. 27, 1906, p. 425.

*A*14—Unpublished 1904 thesis, Cornell University.

*A*15—*Engineering* (London), March 3, 1922, p. 249.

*A*16—Reibung und Schmierung; by L. Gümbel, 1925 edition, p. 109; M. Krayn, Berlin.

*A*17—*Mechanical Engineering*, October, 1927, p. 1109.

*A*18—*Journal of the American Society of Naval Engineers*, November, 1922, p. 564.

*A*19—S.A.E. JOURNAL, July, 1922, p. 49; and TRANSACTIONS, vol. 17, 1922, Part I, p. 215.

*A*20—Friction; by Thomas Ernest Stanton, 1923 edition, p. 109; Longmans, Green & Co.

*A*21—Proceedings of the Institute of Civil Engineers, vol. 226, 1928, Part II, p. 242.

*A*22—Determined from Figs. 1, 2 and 3 in this paper.

preferred to elaborate upon certain of the factors embodied in equation (D1).

Viscosity Determinations

Viscosity, or fluidity, is here taken in poises. For guidance of engineers who are not familiar with this system of centimeter-gram-second units, a simple method is offered for approximating this absolute viscosity value in cases in which this important oil characteristic has not been otherwise more definitely established. Viscosity determinations are generally made dependent upon the rate of orifice flow as measured by Saybolt, Engler or similar instruments, and such viscosimeter readings must be converted into μ values by the use of certain variable multiplifying factors[4], to obtain a true measure of the fluidity of the lubricants.

The unit load generally encountered when working with perfect-film lubrication is so low that no allowance need be made for any minor variation in viscosity that is likely to accompany such limited pressure increase.

Other physical and chemical specifications of any given oil may likewise serve as a measure of its lubricating qualities, but viscosity is a property of primary importance that is intimately associated with fluid shear and may be quantitatively determined as a basis for fixing friction coefficients.

It has been found convenient to resort to an approximate evaluation of viscosity in accordance with the following hyperbolic relation:

$$\mu = \frac{C_3}{T - T_0} \qquad \text{(D2)}$$

While the factor T as used herein may not accurately respond to the oil-film temperature, it is sufficiently close for present purposes since the reported bearing temperatures are usually obtained by placing a thermal element in a drilled hole in close proximity to the rubbing surfaces.

The change in viscosity with temperature actually follows a more complex law than is expressed by equation (D2). A straight-line plot over a wide range, drawn upon logarithmic cross-section paper, leads to more accurate results[5]. According to Prof. G. B. Upton, of Cornell University, each oil group, as derived from a naphthene or a paraffin base, will head toward its own focal point when plotted in this manner.

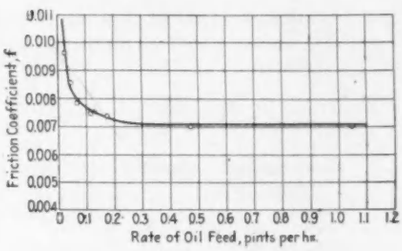


FIG. 5—INFLUENCE OF OIL SUPPLY UPON THE COEFFICIENT OF FRICTION

From Tests Made with Texico Cetus Oil under Conditions of $P=139$ Lb. per Sq. In., $B=158$ Ft. per min., and $T=150$ Deg. Fahr.

The usual range of the given constant C_3, corresponding to some of the more common lubricants other than engine-crankcase oils, as deduced from various sources[6], is presented in Table 2, in which the lubricants have been progressively graded into certain groups according to their fluidity.

TABLE 3—ENGINE-OIL VISCOSITY FACTORS

Constants		Kind of Oil
C_3	T_0 Deg. Fahr.	
9	88	Light engine-oil
15	90	Medium engine-oil
24	92	Heavy engine-oil
30	94	Extra-heavy engine-oil

The underlying relations become more apparent from the graphical representation in Fig. 4, in which the oil viscosity has been charted in accordance with the arbitrary grades stipulated in Table 2. Table 1 also gives the viscosity characteristics for other oils used in the listed tests.

The relative viscosity of an oil can be visualized by means of the chart in Fig. 4, which refers to the ordinary run of machine oils; oils for special purposes might have constants that are distinctly different. For instance, the factor T_0 remains substantially constant in automobile crankcase lubricants; and C_3 becomes the variable, as will be seen from Table 3, which is deduced from papers by G. W. Burke[7] and O. M. Burkhardt[8].

Not all engine oils follow precisely the given values, but Table 3 will afford a basis for judging their characteristic difference as compared with the more common oils cited in Table 2, wherein the constants C_3 and T_0 change in a mutual relationship. It will also be apparent that, for efficient engine lubrication, the engine temperature must be maintained considerably above $T_0 = 90$ deg. fahr.; otherwise the oil viscosity and resulting friction tend to become inordinately high.

Recent literature further introduces the rather indefinite term "oiliness" as an important property of a lubricant. While this measure of adhesion or spread capacity does have a meaning as applied to imperfect metal-to-metal rubbing surfaces, particularly at starting speeds, the present bearing investigation, as directed toward perfect-film lubrication, does not disclose

[4] See Mechanical Engineers' Handbook; by Lionel S. Marks, second edition, pp. 279-280; also Mechanical Engineers' Pocket Book, by William Kent, tenth edition, p. 886.

[5] See Kingsbury, Thrust-Bearing Catalogue No. C (1922 issue), p. 57.

[6] Among such, see W. H. Herschel on, Viscosity and Friction, THE JOURNAL, January, 1922, p. 31; and TRANSACTIONS, vol. 17, 1922, part I, p. 282.

[7] See THE JOURNAL, September, 1926, p. 249; and TRANSACTIONS, vol. 21, 1926, part II, p. 262.

[8] See THE JOURNAL, September, 1926, p. 657; and TRANSACTIONS, vol. 21, 1926, part II, p. 50.

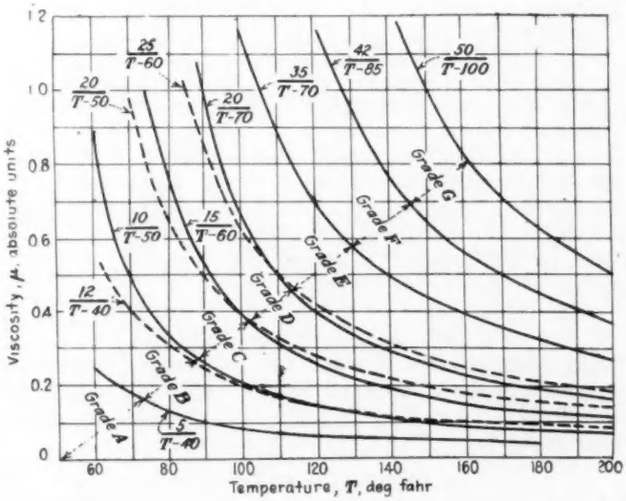


FIG. 4—VISCOSITIES OF VARIOUS GRADES OF OIL

Descriptions of These Grades Are Given in Table 2

TABLE 4—VALUES FOR CIRCUMSTANCE CONSTANT

C_2 [a]	Lubrication	Workmanship	Attendance	Location
1	Oil bath or flooded	High-grade	High-class	Clean and protected
2	Continuous drop-feed	Good	Reasonably good	Ordinary favorable conditions
4	Oil-cup or grease, intermittent	Fair	Poor	Exposed to dirt, grit or other unfavorable conditions

[a] Excessive bearing distortion might double the given C_2 constants.

any such additional oil property except as this may have been inferentially included in the given viscosity factor μ.

Type Constant

For the present, only a brief discussion will be entered into relative to type constant C_1. The given average value of $C_1 = 7.5$ is applicable solely to rigid bearings provided with reasonably stiff rotating journals. Such shafts possess the property of readily pumping the oil thereunder, and this in turn establishes a relatively high lifting pressure for the journal to ride upon, even when running at comparatively low rubbing velocity. A different method for feeding or working the lubricant between the rubbing surfaces is found to modify the resulting friction coefficient.

Circumstance Constant

The quantity of oil supplied to the journal must be allowed for to arrive at a correct appraisal of the coefficient f. The term "flooded supply" indicates a surplus feed. The value of f is reduced to the minimum under these conditions, assuming the oil to be evenly distributed, but bearings generally run on a more stinted supply that is likely to fall far short of the standard set for flooded lubrication. This difference in rate of oil feed finds expression in the circumstance constant C_2, since a cutting down of the rate of oil feed is usually accompanied by a measurable friction increase, especially as the minimum allowable oil need is approached.

Oil supplied at a rate of about 350,000 sq. ft. of surface covered per gallon of lubricant, assuming the shell to contact with only one-half the journal periphery, usually proves sufficient to assure the desired minimum of friction losses in engine bearings. The bearings can be said to be flooded when the oil supply is increased to three or four times this stipulated value; that is, a feed allowance of about 1 gal. for each 100,000 sq. ft. of journal surface, this area being equal to $(\pi/144)\ DLN$.

Restricted oil supply no doubt accounts for much of the erratic nature of the friction-test data found in print. The rate of oil feed should be definitely reported in all cases.

It may be of interest to cite certain tests in which the coefficient f has been determined on the basis of different oil-feed rates. Fig. 5, representing test No. A8, in Table 1, as reported by W. F. Parish, shows the influence of a stinted oil supply. The coefficient begins to rise markedly after the oil supply is cut down to about ¼ pint per hr., which represents a rate close to 400,000 sq. ft. of equivalent rubbing surface covered per gallon of oil in this Thurston double-block test-machine.

When the supply of lubricant is kept below the stipulated critical value of 350,000 sq. ft., the resulting friction coefficient seems, from Fig. 5, to increase inversely as the eighth root of such reduced rate of feed. This deduction is confirmed by oil-endurance tests such as those reported by R. C. Carpenter and Herman Diederichs[9].

Factors other than the abundance of lubricant supply are likely to exert a considerable influence upon friction coefficients. Among such may be cited rigidity of bearing structure, journal fit, workmanship and finish of rubbing surfaces, attendance, location, and similar conditions.

Table 4 is intended as a convenient guide in appraising the composite circumstance constant C_2. It makes allowance for the character of design and machine work, alignment and rigidity of the bearing, and such other structural features as are likely to affect friction losses. Suitable numerical values may be interpolated by modifying the given values to suit any other combination of circumstances. Bearings lacking in rigidity or failing to maintain a uniform distribution of lubricant or load, eccentric straps, for instance, might require double the prescribed C_2 constants.

The values listed in Table 4 apply to journals loaded on one side only and unrestrainedly mounted to lift away from the shell; when the journal is confined between oppositely disposed blocks, the coefficient f will usually be increased, because of the less favorable conditions for building up the oil film.

[9] See Experimental Engineering, fifth edition, p. 203, J. Wiley & Sons, New York City.

TABLE 5—MACHINE FITS

Class of Fit		Fit Constant $1/K =$	Fit Constant $K =$
A	Precision	0.0007	1430
B	Close running	0.0014	715
C	Slow running	0.0021	476
D	Fast running	0.0031	323
E	High speed	0.0042	238
F	Loose	0.0063	159
G	Lateral or end	0.0125	80

TABLE 6—BEARING FITS

Class	Kind of Service
A	Automobile piston-pins, non-adjustable
B	Automobile-crankshaft bearings
C	Slow-speed-engine main bearings
D	High-speed-engine main bearings Slow-speed-engine outboard bearings Vertical electric-motor bearings Spur-gear journals, fluctuating load Line-shaft and pulley bearings
E	Horizontal steam-turbine bearings Horizontal electric-motor bearings Spur-gear journals, ordinary service
F	Bending-roll bearings
G	Normal end-play for bearing shoulders Quarter-box-bearing bore

Effect of Journal Fit

The snugness of a journal fit also exerts an important influence upon the coefficient of friction. Characteristic machine fits ordinarily used in shop practice

are classified in Table 5, in which the basis of the fit allowance is:

$$\delta = \sqrt{D} \div K \qquad \text{(D3)}$$

The stipulated fits usually are held to a tolerance of $\pm\delta/6$, and it will be seen that the given values of K are commonly stepped off in a ratio of about $1\frac{1}{2}$:1 or 2:1 in machine-shop practice.

Table 6 shows the kinds of fit that are suited to different purposes. Piston-pins are frequently fitted one class closer than the corresponding crankpin and main-bearing fits, and non-adjustable parts are usually made one or two fits closer than corresponding parts that are split and made adjustable. A reciprocative load is made less than a steady-journal load.

Slow speed bearings are usually fitted closer than high-speed bearings, but the Class-C fit is the closest that is desirable for a free-running bearing, to allow sufficient clearance for turning freely and readily establishing a perfect oil-film. A cap clearance of at least 0.002 in. seems to be necessary in split journals whose shells are bolted together. Furthermore, assuming the shell housing to remain relatively cool, a localized heating on part of the journal to an extent of 150 deg. fahr. would cause an expansion of about 0.001 in. per inch of journal diameter, and hence would be likely to bring about seizure in a closely fitted shell.

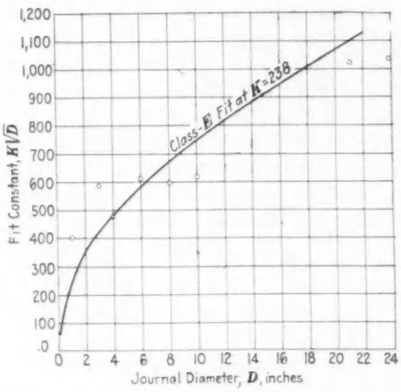


FIG. 6—FITS FOR HIGH-SPEED BEARINGS

Points Plotted Are Taken from Practice

In Fig. 6 is plotted a curve showing the recommended Class-E fit as applied to horizontal electric-motor bearings; and there is also indicated the comparative practice of the General Electric Co.[10], which fits the smaller diameters more snugly than is dictated by equation (D3), while the larger journals are made somewhat looser, probably because they afford an inherently greater contact area, S, for a given fit, K.

Journal-Contact Factors

The $K\sqrt{D}$ factor is found to be one of the more important constants required to bring into agreement the diverse equations listed in Table 1. In accordance with equation (D3), $K\sqrt{D}$ may also be written D/δ. This ratio of fit to diameter plays a significant part in the present investigation, since the arc of effective film contact does not usually extend over the whole of the projected bearing area.

When the unit-pressure basis is taken as defined for equation (D1), the pressure intensity P/S will actually be carried upon an effective or net contacting journal surface equal to its total projected area multiplied by the factor S. Furthermore, only such portion of the lubricant as has been squeezed down into an exceedingly thin film is capable of floatingly sustaining any considerable journal load. These considerations make apparent why the fit ratio D/δ to a large extent controls the contact factor S. As determined from the limited available data, the numerical value for this factor may be estimated thus:

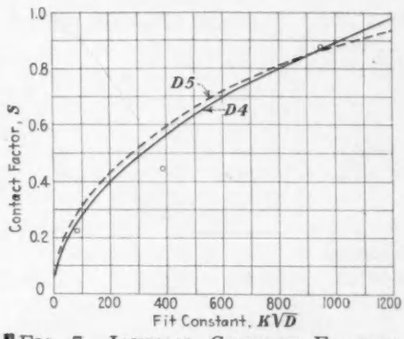


FIG. 7—JOURNAL CONTACT FACTORS ACCORDING TO EQUATIONS (D4) AND (D5)

$$S = \sqrt{\left(\frac{K\sqrt{D}}{1250}\right)} \qquad \text{(D4)}$$

Equation (D4) shows the factor S to depend primarily upon the fit factor $K\sqrt{D}$. Should the sides of the bearing shell be cut down to a chord length less than that corresponding to the value of S, then the entire restricted shell area may be considered as having been brought into effective contact.

In Fig. 7 is plotted equation (D4), together with certain check points as derived from the Stribeck tests run at $V = 138$ ft. per min. A dotted reference curve has also been added, as deduced upon the following trigonometrical basis:

In Fig. 8, the journal is assumed to be motionless and resting in substantial line contact upon the bottom of the shell; hence, the radial oil-film thickness β, measured for the angular offset as taken to either side of such center line, may be estimated as follows:

$$\beta = \frac{\delta}{2}(1 - \cos\alpha) = \frac{\sqrt{D}}{2K}(1 - \cos\alpha) \qquad \text{(D5)}$$

The value of the contact factor S then becomes equal to $\sin\alpha$, and the cited reference curve has been determined by giving to β a fixed value of about 0.0007 in. and to α a value of 59 deg., as applied to the Stribeck test No. A.5, Table 1.

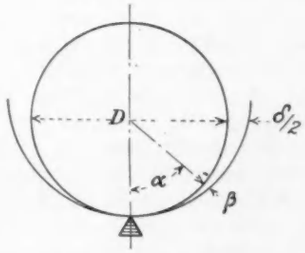


FIG. 8—NOTATION FOR THICKNESS OF OIL FILM

The present findings do not point to any need for altering the journal fit in accordance with the viscosity of the lubricant, assuming that the film is kept sufficiently thick to floatingly sustain its load.

It would unduly complicate this subject to attempt further to correct the β determination for bodily lift and eccentric tilt of the rotating journal. Experimental work has been done along these lines by Soney and Boswell, A 15 in Table 1, from which the following relation has been transposed:

$$l = \frac{\sqrt{\mu}}{2200}\left(\frac{V}{P}\right)^{\frac{1}{4}} \qquad \text{(D6)}$$

[10] See Mechanical Engineers' Handbook, by Lionel S. Marks, second edition, p. 754.

TABLE 7—PRESSURE COEFFICIENT n_0

Test Reference	Experimenter	Value of n_0	Constant C_2
A1	Tower	About 1	About 1
A2	Tower	0.7	1.7
A4, A5	Stribeck	0.73	1.0
A6	Lasche	1 or less	1.0
A21	Goodman	0.84	1.0
A7	Thomas and others	5/8	1.0
	Ring-oiled-bearing practice	2/3 or less	1.0 to 1.25
Equation (D7)	McKee	0.45	1.5

A relatively low degree of contact is invariably found in checking back on loosely fitted journals, which affects the numerical value of the factor S. This seems generally not to exceed 0.80 in commercial practice, although it may more closely approach unity with special care, as in laboratory tests.

Maximum-Contact Factors

The limiting value assumed by the factor S has a further significance which might be better brought out after discussing some characteristic tests as reported by S. A. and T. R. McKee[11]. These tests, made upon a 1¼-in. connecting-rod bearing, were especially designed to show the effect of journal fit upon the friction coefficient f. In this series of determinations, not only was the shaft length L varied in relation to its diameter D, but the fit of journal experimented with was given different δ values. Unfortunately, the ZN/P or grouped method of presentation was resorted to, except for one typical set of data. The experimental data were not reported in full but in terms of a combined relation, in which Z is taken as proportional to the given viscosity μ. Because of this, it has been impossible here to utilize these tests to their full extent.

When the coefficients from the McKee graphs are plotted on a K basis for $ZN/P = 400$, as in Fig. 9, this reveals that all the given L/D ratios show substantially the same f values except when the L/D ratio is reduced to less than ¾. For ordinary journal proportions, where $L/D > 1$, end leakage therefore should not appreciably modify the resulting friction coefficient. A similar conclusion is deducible from Fig. 9 of the cited McKee reference. The list in Table 1 likewise indicates a rather wide change in the L/D relation, but no noticeable difference in f values has been detected because of such difference.

The heavily drawn straight line in Fig. 9 shows the McKee friction coefficient to increase directly with the fit factor K. Furthermore, as this straight line approaches the zero value of K, the curve is deflected, as shown to enlarged scale in Fig. 10. Below the respective critical points marked K_0, the plot responds to normal or unconstrained film friction, while above K_0 the journal fit evidently runs too close to permit freely establishing an oil film. The resulting friction coefficient for values of K lying above K_0 may be roughly expressed by the equation

$$f_k = \frac{K\sqrt{D}}{75{,}000} \times \frac{\mu V}{P} \qquad \text{(D7)}$$

Attention is directed to the point of tangency, designated as K_0 in Fig. 10, which shows a low average value of about 500; that is, $K_0\sqrt{D} = 560$ below a velocity of 180 ft. per min. The circumstance constant C_2 can be only inconclusively evaluated from the given data, but seems to be about 3/2. Referring now to equation (D4), the resulting contact factor S for the given K_0 value is evidently very low and indicates that only a little more than one-half of the full projected journal area is brought into effective service, after which the friction coefficient changes over to the higher f_k value defined by equation (D7).

Such a shift seems to be characteristic of all the tests listed in Table 1 for which sufficient data are available for analysis. In some of these tests, the constant $K_0\sqrt{D}$ lies closer to its ultimate value, 1250, as fixed by equation (D4). For the Stribeck tests, for instance, it may be rather definitely determined at $K_0\sqrt{D} \approx 910$, while for the Tower test, (A1), $K_0\sqrt{D} \approx 1100$. In the case of commercial high-speed bearings, the value of $K_0\sqrt{D}$ rarely exceeds 850. Accordingly, the maximum of contact area that can be realized before the cited change in friction behavior occurs may be written:

$$S_0 = \sqrt{\left(\frac{K_0\sqrt{D}}{1250}\right)} \qquad \text{(D8)}$$

Journal-Pressure Exponent

It is further found that the factor S_0 is closely associated with the maximum value that can be reached under given conditions by the pressure exponent n, the relationship being expressed as follows:

$$n_0 = S_0^2 = \frac{K_0\sqrt{D}}{1250} \qquad \text{(D9)}$$

The factor n_0 is an inherent characteristic of any given bearing, depending upon the maximum contact area attainable, since not all of the projected journal area can ordinarily be made effective. Consideration must be given to bearing peculiarities that are rather difficult to appraise definitely in individual cases, but which need to be taken into account, because of shaft flexure and other inherent causes, to arrive at a correlated result.

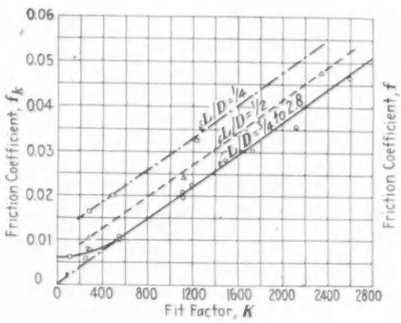


FIG. 9—CHANGE OF FRICTION WITH FIT

Curve Plotted for the Condition of $ZN/P = 400$

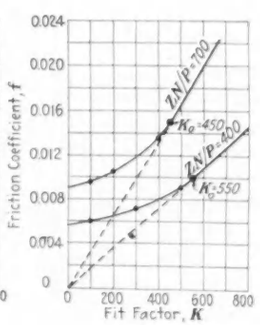


FIG. 10—CURVES FOR FRICTION OF UNCONSTRAINED FILM

From the foregoing it may be concluded that the pressure exponent n cannot exceed the ascribed value

[11] See Transactions of the American Society of Mechanical Engineers, 1928 Annual Meeting Paper entitled, Friction of Journal Bearings, etc.

of n_0. It is found that the desired coordination of pressure exponents when $K<K_0$ is expressed by the relation

$$\frac{n}{n_0} = \left(\frac{S}{S_0}\right)^{\frac{1}{4}} = \left(\frac{K\sqrt{D}}{1250 n_0}\right)^{\frac{1}{4}} \qquad \text{(D10)}$$

By transposition

$$S_0^2 = n_0 = \left(\frac{1250}{K\sqrt{D}}\right)^{\frac{1}{3}} n^{\frac{2}{3}} \qquad \text{(D11)}$$

The value of $K\sqrt{D}$ and the pressure coefficient n are generally known for any given test conditions. The

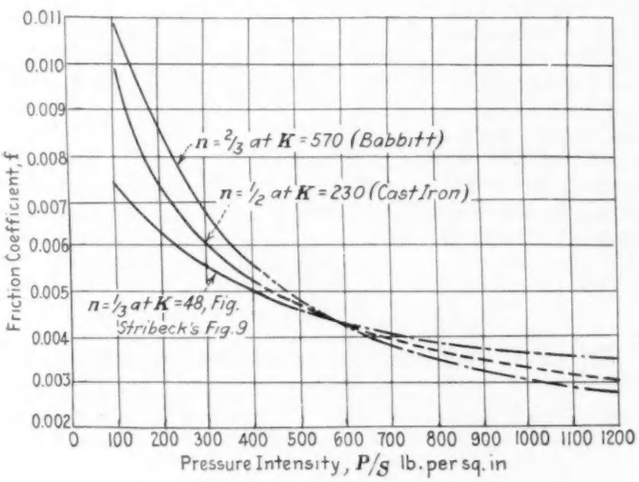


FIG. 11—STRIBECK DATA ON THE BASIS OF P/S

constant n_0, which represents a measure of the inherent peculiarities of the bearing, can then be determined by the use of equation (D11). The foregoing formulas are based upon data from tests in which a substantially unbroken film is maintained and the contact factor S is kept less than the available shell periphery corresponding to S_0.

The n/n_0 ratio tends to approach unity with closer journal fit. The limiting pressure coefficient n_0, as applied to any given bearing, may also be evaluated by charting a series of tests that have been run under fixed V and μ conditions and then finding the critical point K_0 at which the coefficient n ceases to respond to a further reduction in the journal fit.

The value of n_0 appears to vary within rather narrow limits; in the absence of a better guide, n_0 can be interpolated from the comparative values given in Table 7, derived from those of the listed tests for which the necessary data were reported.

This simple empirical treatment offers an important advantage, because it can readily be applied directly to solving certain of the more common fluid-friction problems without resort to intricate correction to make the results conform to expectations. A simplified exposition of the prevailing theory of lubrication, initiated by Osborne Reynolds, as based upon journal lift and its eccentricity within the shell bore, has been given[12].

In practice, the value of n_0 as applied to the oil-ring type of bearing seldom exceeds 2/3 and may be ½ or less. This means that only a limited portion of the projected journal area is ordinarily brought into effective contact, but it does not imply that the accompanying friction coefficient takes on an abnormally high value.

The relationship of the friction coefficient with reduced values in the pressure exponent n is presented in Fig. 11, in which the Stribeck-test coefficients have been plotted upon a P/S or actual-pressure basis. Equations have been used to extrapolate the curves beyond the range of the tests. The lowest curve, from Fig. 9 of Stribeck's report, is for an exceptionally loose journal fit corresponding to $K = 48$. All three curves cross at an actual pressure of about 600 lb. per sq. in. Below this intensity, the loosely fitted journal really runs with an f value or frictional drag that is somewhat lower than for the more tightly fitted journals.

Fig. 11 indicates that more favorable results are likely, under moderate unit pressures, when a journal is not too closely fitted. The higher pressure intensity P/S tends to reduce frictional loss, provided that the critical pressure P_c is not exceeded. The use of the looser fit involves a lowered total load-carrying capacity for a given journal, but the accompanying friction reduction is usually sufficient to compensate for the larger bearing area required for a given load.

Under conditions of substantially perfect film formation, the pressure exponent n may ordinarily be expected to drop with a reduced fit-constant K, in accordance with equation (D10). A value of n materially lower than that prescribed indicates incipient metallic contact between the rubbing surfaces and a partial breaking down in the oil-film continuity.

Journal Reference Equation

Equations (A1) from Tower and (A6) from Lasche most nearly represent perfect tests for film lubrication. They can be brought to a common basis, when $P < P_c$, by the following equation:

$$f_1 = \frac{1}{7.5} \cdot \frac{\sqrt{(\mu V^{2/3})}}{P/S} \qquad \text{(D12)}$$

This equation measures the minimum film friction that is likely to be realized under given operating conditions and therefore serves as the desired criterion with which to compare other oil-borne-journal friction coefficients. A reference standard of this kind also proves helpful in the analytical treatment of more complex friction coefficients.

Equation (D12) shows the coefficient f_1 to be independent of the total load carried by the bearing. To obtain the required unity value for the pressure coefficient n, the fit factor $K\sqrt{D}$, of equation (D10), must closely approach its limiting value of 1250. Such snug-fit conditions do not generally prevail in commercial practice; instead, it is preferred to fit fast-running journals more loosely, and this in turn is found to change the character of the resulting friction and cause the frictional drag to increase with the augmented pressure intensity. The expectations under such modified fit conditions will now be developed.

Coordination of Friction Coefficients

The exponential values given in the equations for friction coefficient in Table 1 were largely fixed by plots and in part upon judgment to make them reasonably consistent with the reported test results and were rounded off to the nearest simple fraction. The available data were in some cases inadequate to establish

[12] See *Engineering*, (London), July 30, 1915, p. 101; and Aug. 13, 1915, p. 154.

satisfactory exponents, but it is believed that the averaged values should eliminate any serious discrepancy.

The sought-for interrelation between the various friction coefficients applicable to journals running on a substantially unbroken oil-film is found to depend largely upon the pressure exponent n. On the basis of equation (D12) and assuming $P<P_c$, the analyzed results presented in Table 1 may be empirically combined in the following manner:

$$f_c = \frac{C_2 n^3}{7.5} \cdot \frac{\sqrt{(\mu V^{2/3})}}{(P/S)^n} \qquad \text{(D13)}$$

The given f_c values are such as are usually obtained in the best bearing-practice. A curve corresponding to equation (D13) is plotted in Fig. 12, with certain check points indicated.

In cases in which the fit factor $K\sqrt{D}$ is carried beyond its normal value $K_0\sqrt{D}$, the resulting friction, in accordance with the findings embodied in equation (D7), will be increased over the given f_c value in the ratio of K/K_0. Equation (D13) serves to coordinate the listed journal tests and makes possible the predetermination of friction coefficients under different operating conditions. Certain practical applications of the foregoing deductions will be discussed later.

Journal-Test Discussion

The findings of certain of the tests listed in Table 1 will be briefly reviewed in consecutive order.

A1, Tower.—These classic experiments are of the oil-bath type in which a 3.9-in.-wide bearing-block was disposed upon the top of a 4-in.-dia. journal. An oil-filled pan was mounted directly underneath the journal and served to keep the rubbing surfaces flooded. Quite a variety of well-known lubricants were used, including sperm, lard, rape, olive, and mineral oils, also mineral grease. No viscosity determinations are given, but the viscosities can be estimated with a fair degree of accuracy. The so-called grease appears to be a fairly heavy oil having a viscosity of about $\mu = 20/(T-70)$, which would hardly be designated as grease now. All the cited tests lend themselves to consistent plots except that of rape-oil, which runs at about 80 per cent of the f value given by the equation (A1).

The fit constant was not reported but appears to lie close to $K = 570$, or $K\sqrt{D} = 1140$, when allowance is made for wear during this extended experimental work. The value of n closely approaches unity and shows a high degree of contact. The contact factor S works out to about 96 per cent, according to equation (D4) with $C_2 = 1$, indicating that substantially all of the available chord length of 97 per cent was effective. Fig. 13 presents an experimental determination of the measured pressure distribution for the Tower journal running at $V = 157$ ft. per min. and $P = 342$ lb. per sq. in. A sine-shaped pressure distribution of this kind fully spans the effective contact portion of the journal, the curve usually being somewhat warped in the direction of journal movement.

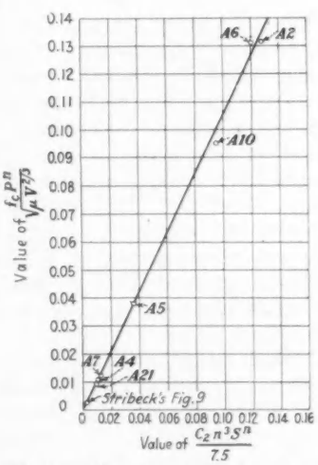


FIG. 12—COMPARISON OF EQUATION (D13) WITH EXPERIMENTAL DATA

For This Comparison, Equation (D13) Is Transposed so that the Expression Given for the Scale of Ordinates Becomes Equal to the Expression for Abscissas Above

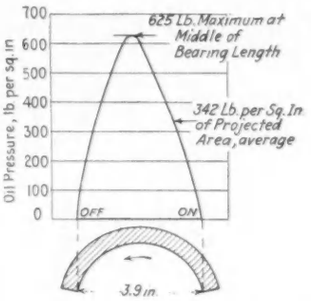


FIG. 13—OIL-PRESSURE MEASUREMENTS IN TOWER TEST *A1*

The Journal Used in This Test Was 4 In. in Diameter and 6 In. Long

A still slower-speed run was made in connection with this test, using a mineral-oil bath held at 90 deg. fahr. with $\mu \approx 0.45$ without materially changing the cited pressure distribution. The resulting friction coefficient at $V = 21$ ft. per min. may be expressed as

$$f = \frac{0.13}{P^{3/4}} \qquad \text{(D14)}$$

The pressure exponent is reduced to $n = 3/4$, showing deficient film formation, while the value of f for an average P of 220 lb. per sq. in., is now found to be approximately three times that of equation (A1). At a rubbing velocity of somewhat less than 200 ft. per min., the f value given by equation (A1) becomes equal to that of equation (D14), which shows that a velocity of the given magnitude is needed to floatingly sustain the stipulated unit load. The oil bath used was exceptionally favorable to film formation at low rubbing velocities. The friction coefficient would ordinarily assume a much higher relative value when a journal is running so near to the zero starting speed.

A2, Tower.—A siphon lubricator was employed in these tests instead of an oil bath. This presumably was equivalent to a slow drop-feed, but nothing concrete is cited as to the rate of oil supply. The fit constant has been assumed to be equal to that of test *A1*, since apparently the same journal was used in this experiment.

On the basis of equations (D10) and (D12), n_0 seems to be slightly greater than $n = 2/3$, and the corresponding circumstance constant C_2 is about 1.7. This check is complicated by the fact that the estimated arc of contact S_0 is somewhat greater than the actual span length that was given to this restricted bearing block, as indicated in the column of remarks in Table 1.

A3, Goodman.—This test was a repetition of Tower's test No. A1 and shows substantially identical results. In addition, the projected width of the bearing block was progressively reduced from full width to approximately one-fourth width and kept at fixed unit load without causing any appreciable difference in the accompanying friction coefficient. This finding is thought to justify taking the friction coefficient in equation (D1) as inversely proportional to P/S.

A4 and A5, Stribeck.—The first cited of these tests was made upon a line-shaft bearing of the ring-oiler type. The shell was of cast iron, bored to a comparatively loose Class-E fit ($K \approx 230$), and this was thor-

oughly worn-in prior to running the experiments. The *A*5 test was made upon a babbitt-lined bearing of similar type but more closely fitted ($K = 570$) which was apparently about the limit of snugness to allow good formation of film under the conditions. It has been necessary to estimate the respective δ values, since Stribeck does not give his actual fit allowances but reports only differences in journal clearances.

The two tests analyzed show no important difference in friction coefficients under substantially identical P/S unit-pressure conditions. Instead of taking the f equation, given in the form as No. A5, the results of these tests with more closely fitted bearings may also be approximately expressed by increasing the 0.004 constant of equation (A4) to about 0.0058. This difference in friction coefficients may be ascribed to the considerably higher pressure-intensity that prevails in the case of the looser journal, which actual pressure appears to be about twice as high as in the more closely fitted journal. In accordance with equation (D4), the contact factors work out in these cases to about 9/16 for the cast-iron-shell fit and about 7/8 for the babbitt-shell fit. While these estimated values do not check closely with the foregoing coefficient-ratio findings, the discrepancy is thought to lie in an incorrect assumption for the fit constant K as taken for the looser journal.

This inference is corroborated when considering a still looser cast-iron bearing that Stribeck tested and reported upon in his previously cited Fig. 9, which is reproduced in Fig. 14. The curve marked $K = 230$ is representative of test No. A4 but it will be observed that the curve marked $K = 48$ shows not only the decided effect of abnormal journal clearance but also a lowered friction coefficient for any given pressure $P < P_c$. The Stribeck Fig.-9 test was run at $\mu = 1.95$ and $V = 138$ ft. per min., and the corresponding equation may be taken as

$$f = \frac{0.021}{P^{1/2}} \qquad \text{(D15)}$$

The relation shows that the pressure exponent n does drop substantially when the contact factor is reduced. In accordance with equation (D4), S is about 1/4, which seems to be consistent with other checks for its value.

*A*6, *Lasche.*—This series of tests was run upon a large high-speed-turbine or motor type of ring-oiled bearing. The fit constant, while not directly stipulated, is estimated at $K = 320$ (Class D), and the corresponding contact factor S at about 0.91. The journal in question appears therefore to have been fitted more closely than Table 6 indicates, but this is characteristic of laboratory tests.

The value of the pressure exponent n of this run lies near to unity, which corroborates the Tower findings and shows that a substantially perfect film must have been maintained also in the Lasche tests. The value of μ could not be determined definitely from the given test data, but apparently a relatively light grade-C mineral-oil was used.

*A*7, *Thomas and Others.*—This run was conducted upon line-shaft bearings of the ring-oiled type, for which n_0 apparently assumes a value of 5/8 at $K\sqrt{D} = 370$. While the viscosity is given, this has been reported in arbitrary rather than absolute units, and its value therefore had to be interpolated. An outstanding feature of the Thomas test is that the results closely correspond with those attained in the best commercial practice when $C_2 = 1$.

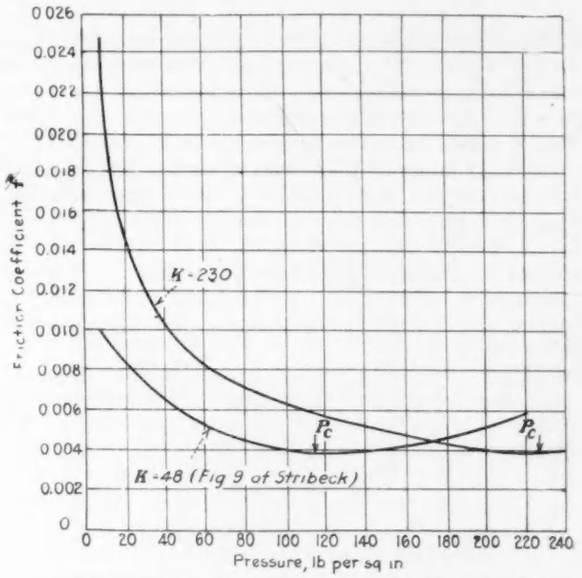


FIG. 14—TEST DATA FROM STRIBECK'S FIG. 9

*A*8, *Parish.*—Certain aspects of this report have already been touched upon. These tests were made upon a Thurston-type friction machine and show a low pressure-exponent of $n = 1/3$, because of imperfect film formation. As is more fully pointed out in discussing test A9, immediately following, the use of such oppositely mounted thrust-blocks proves inefficient, because the lubricant is not evenly supplied to both complementary surfaces. These conditions are unsuitable for comparing the lubricating performance of different oils to the best advantage.

*A*9, *Hopps.*—The journal was run vertically in this test, with snugly fitted complementary thrust-blocks on each side and the entire bearing immersed in a bath of oil. The arrangement is similar to that of a Thurston testing-machine except for a more ample provision for uniform distribution of the lubricant. The fact that equation (A9) shows a pressure exponent as low as $n = 1/2$ points to an unfavorable film formation whenever two such oppositely disposed bearing-blocks, mounted so that they cannot tilt, are spring-pressed toward the shaft. This deduction is confirmed by the further fact that different combinations of rubbing materials produce different friction coefficients, thus showing that this test has to do with partial rather than perfect-film lubrication. Under these conditions, unhardened wrought-iron or mild-steel journals, running on bronze, reveal a friction loss 1/4 greater than when the same journal surface is hardened.

*A*10, *Heinmann.*—This report was based upon the use of a special testing-machine provided with a heavy rotor shaft in which the friction coefficient was determined by measuring the deceleration. The more consistent results were obtained with a stipulated slow-running Class-C fit of $K \approx 450$. The indications are that this type of machine is not a satisfactory means for finding friction coefficients, since the accompanying circumstance constant appears to be $C_2 \approx 2.2$.

*A*11, *Martins.*—This early investigator reported viscosity in absolute units and was probably the first to

recognize the scientific importance of properly fixing upon its value. The pressure exponent is very low, which indicates that perfect film formation was hardly approached in these experiments.

*A*13, *Kingsbury.*—The apparatus used in this test is exceptionally large and was run at high rubbing speeds. The loads carried were relatively heavy and apparently caused the long journals to spring perceptibly. This tends to modify the resulting friction coefficient. The appraisal is further complicated because of the different classes of fit used for the large and small journals and the failure to segregate the corresponding friction losses.

Equation (A13) indicates a combined friction coefficient, for the three tandem bearings tested, about 45 per cent of that indicated by equation (D11). The small effective-contact area is no doubt the explanation of so low a result. The fit factor for the 9-in. journal is $K \approx 400$ and, for the larger journal, $K \approx 130$, thus giving an average value of $K = 265$, which is a little closer than the prescribed Class-F fit.

The corresponding weighted $K\sqrt{D}$ value is about 970, which, according to equation (D4), should possess an average contact factor of $S \approx 7/8$, while its actual value appears to be only one-half as large. A check indicates a shaft deflection, under the given load conditions, of approximately 0.005 in. for a 40-in.-long bearing, and this is believed to account for a concentration of load near the bearing ends. Judging from the unity pressure exponent, this localized pressure was still carried upon a perfect oil-film without substantial metallic contact.

*A*14, *Woods and Others.*—This series of high-speed tests was made upon a rebuilt watercooled Thurston testing-machine. It will be noted that the pressure exponent is again low, at $n = 1/3$, also that the coefficient increased rather slowly with speed. Under augmented pressure, the plotted f values in Fig. 3 cross the composite value labeled f_j, and thus show signs of film breakdown.

*A*15, *Stoney and Others.*—This test was run upon a special friction-machine using a horizontal journal immersed in an oil bath. Complementary brasses were spring-pressed against opposite journal faces, and they were tiltably mounted to facilitate the formation of a wedge-shaped film. The brasses are described as closely scrape-fitted, but the tiltable blocks in this instance must have adjusted themselves to inclined positions at the equivalent of a Class-E fit at $C = 223$. Taken on this basis, the friction coefficient as determined by equation (D12), with $n = 5/8$ and $C_2 = 1.0$, would still be about 1/8 higher than that corresponding to equation (A15), thus indicating that the assumed n exponent is probably somewhat too high.

*A*17, *Nielson and Others.*—This citation is of interest chiefly because the lubricating oil used was distilled from coal. The fit is loose, at $K = 156$; and, according to equation (D11), n_0 works out to about 3/4. The given friction coefficient appears to be lower than that determined by equation (D13), but this minor difference may be due to an inaccurate appraisal of the pressure exponent n.

*A*18, *Howarth and Others.*—This test was run on a Kingsbury marine thrust-bearing, provided with plural thrust-blocks tiltably mounted to facilitate the formation of wedge-shaped oil-films beween themselves and the flat faces of the collars. Equation (A18) shows an equivalent pressure exponent of $n = 2/3$, while the μ and V exponents coincide with equation (D12). Fig. 2 further shows this Kingsbury bearing to operate with a friction coefficient slightly lower than the given oil-borne composite f_j value, and hence the constant C_2 may safely be fixed at 1.0. The equivalent contact factor S, as determined from equation (D4), is approximately 9/16, while the corresponding K value appears to be about 125. This bearing must be effectively tilted, as it operates on a fit factor equivalent to Class-F, which is one step looser than was found for the Stoney test, No. A15. The present analysis demonstrates that, when such tiltable blocks are made to operate with a wedge-shaped film, the results conform closely to the best attained with rotating journals.

*A*19, *Hersey.*—The data for this equation were based upon the coordinated graph presented in a paper by R. E. Wilson and D. P. Barnard, 4th, in which the cited ZN/P relation is used throughout. In this paper[13] a series of recommended ZN/P constants ranging between 5 and 200 is listed as applicable to a fit of $\delta = D/1000$ or $K\sqrt{D} = 1000$. The necessity for such wide variation in this basic relation indicates a failure of the cited method to get at the root of friction phenomena.

Furthermore, while many tests can be plotted on the basis of a straight line for ZN/P, the resulting line seldom heads directly toward the zero axis; and it therefore shows no direct proportionality between f and ZN/P but indicates a root value for the previously defined m, n and r exponents. Comparison of these findings with equation (D13) has been impossible, because the cited grouped factors of the Hersey test do not make possible individual valuation of the pressure exponent n.

*A*20, *Stanton.*—In this interesting test, a bearing bushing surrounded the rotating journal loosely, leaving an eccentric-shaped clearance space and causing the formation of an effective oil-film in only a small contact region, measured peripherally. The bushing was mounted freely so that the rubbing surfaces could adjust themselves, and the entire bearing was run in an oil bath under heavy concentrated loads. Pressure graphs similar to that in Fig. 13 indicate a 5 to 10-deg. lag between the beginning of film contact and any substantial rise in the resulting sine-pressure curve.

The exponent n approaches a zero value in equation (A20). The given f values probably represent the lowest friction coefficients that are likely to be attained under the stipulated conditions, the bearing pressure having reached a critical point at which it is about to go over into partial-film lubrication. Stanton worked with two different fits; namely, one having $K = 50$ and the other even lower at $K \approx 17$. These extremely loose fits no doubt account for the line-like-contact results attained.

In the tests run with $K = 50$, two different kinds of oil were used; namely, sperm-oil, for which $\mu = 0.25$, and rape oil, at $\mu = 0.54$. The total load carried upon the bearing was in all cases kept at $P = 277$ lb. per sq. in. of projected area.

The corresponding pressure exponent n is estimated at about 1/4. In accordance with equation (D13), the resulting friction coefficient for sperm-oil works out to $f_c = 0.0011$, which is about 1/2 higher than its ob-

[13] See THE JOURNAL, July, 1922, p. 59; and TRANSACTIONS, vol. 17, 1922, part I, p. 237.

served value. It seems, however, that this particular observation might have been read too low. As applied to rape-oil under similar conditions, assuming $C_2 = 1$, the value of f_c is found equal to about 0.0016, which is a little lower than the observed value of 0.0017.

In the test under discussion, the film pressure distributed itself over something like a 30-deg. arc, corresponding to a value of $S = 0.26$ in.; but, as shown by the pressure graph accompanying the Stanton report, only about two-thirds of this angular contact area was effectively utilized. The estimated contact value, as appraised by equation (D4), is $S = 0.2$ in. The value of n_0 appears to be about ½, and the average pressure intensity P/S was carried close to 1400 lb. per sq. in.

The journal tests using a still looser fit, $K = 17$, were run by using two relatively heavy oils whose viscosities were $\mu = 1.43$ and 2.09, respectively. The gross film width was reduced in these tests to ⅛ in., which in this instance is found to be just about equal to the estimated S value. The corresponding pressure intensity is thereby raised to about 2300 lb. per sq. in., and evidently is carried beyond the allowable P_c value.

Under these last-named conditions, the calculated f_c coefficient drops below the observed test value unless the pressure exponent is kept at $n = ¼$. A factor that might to some extent reconcile the indicated discrepancy is an allowance for wear on part of the heavily loaded bearing. In this particular line-like-contact region, extreme accuracy in measuring the prevailing fit relations becomes of considerable importance.

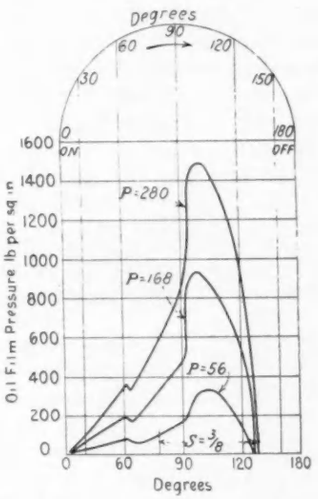


FIG. 15—PRESSURE DISTRIBUTION IN GOODMAN TESTS

Mean Pressure P, in Lb. per Sq. In., Is Indicated for Each Curve. Measurements Were Made at Each 5 Deg. by Rotating the Bushing

The Stanton tests can be most nearly formulated by an equation such as (A20), in which the exponent n is kept fixed at ¼. Further reduction in the fit K apparently is not effective in altering the given pressure exponent. This aspect seems the more probable when considered in the light of the author's previously cited High-Pressure-Bearing Research, which finds the friction coefficient pertaining to such partial-film lubrication best approximated by the use of a negative pressure exponent having a value of $n = -¼$.

A21, Goodman.—This test was conducted with a special friction machine using a full semicircular shell superimposed upon a journal that dipped into an oil bath as in the Tower tests. The fit was made rather loose, at $K = 70$ or $K\sqrt{D} = 171$. The corresponding pressure exponent is about $n = ½$, and the value of $K_0\sqrt{D}$ seems to be about 1050.

The actual pressure distribution, as determined under different loads is recorded in Fig. 15. For some reason the pressure on the "off" side of the shell drops abruptly, and the film pressure ceases at about 50 deg. inward from the off-side edge. The cited peculiarity may be due to excessive end-leakage, since the journal length L is only one-half the diameter D.

The estimated contact factor, in accordance with equation (D4), is $S = 0.37$. Taking $n = ½$ and $C_2 = 1$, the expected coefficient of friction, as determined from equation (D13), becomes

$$f_c = \frac{1}{100}\sqrt{\left(\frac{\mu V^{2/3}}{P}\right)} \qquad \text{(D16)}$$

The Goodman reference equation (A21), in Table 1, agrees fairly well on the whole with this deduction and therefore shows no appreciable rise in friction because of the cited small L/D ratio or the use of a shell made to embrace 180 deg. of the shaft periphery. In view of the low estimated S value, it is not apparent that any materially different results would have been attained by cutting back and relieving the opposite shell sides.

Professor Goodman also made some rather extended comparisons between his observations and those corresponding to the current hydro-dynamic theory of film lubrication, without, however, finding a substantial agreement therewith except under light loads. As a consequence, considerable controversy arose. Some of those taking part in the discussion held that, owing to such divergence, the present theory might be faulty or incomplete and not applicable to loosely fitted journals.

One of the commentators regretted the failure of the paper in question to attain any useful or informative results. Such unwarranted criticism reveals a lack of understanding as to what constitutes the distinguishing feature of film lubrication. It is believed that the discussion of equation (D1) in this paper represents a consistent formulation of the point at issue, since the presence of substantial film continuity can be detected as long as the pressure coefficient n does not become negative.

That the journal in the tests under consideration did ride upon a relatively thick oil-film was proved by actual measurement. The determined vertical-lift component of the journal may be approximately represented by the following equation:

$$l_1 = \frac{V^{5/8}}{2300}\sqrt{\left(\frac{\mu}{P}\right)} \qquad \text{(D17)}$$

The horizontal-shift component was found to be so much smaller as to be negligible for our present purposes. Attention is directed to the similarity between equation (D17) and (A21), from which it appears that the coefficient f is about $4.2l_1$. A like relationship is not so evident in the case of equations (D6) and (A15). Lift l_1 is about $3/2.l$ to $2l$, as derived from equation (D6), but both determinations show an oil-film thickness of several thousandths of an inch at $V = 1000$ ft. per min. when the bearing is carrying a moderate unit pressure.

A22, Composite Values.—The composite friction coefficient f_f merely means that, under average laboratory conditions, a coefficient of the given magnitude may be expected. Judged by equation (D13), the corresponding value of S is 0.68. This method of approach also affords a suitable basis for fixing upon the most probable value for the exponents m and r.

To make this paper useful for bearing design, certain

TABLE 8—FRICTION COEFFICIENTS FOR $n_o = 0.64$ AND $P < P_c$

Line	$K\sqrt{D}$	n	S	Coefficient of Friction	Frictional Drag
1	1250	1.0	1.0	$f_1 = \frac{C_2}{7.5}\frac{\sqrt{(\mu V^{2/3})}}{P}$	$f_1 P = \frac{C_2}{7.5}\sqrt{(\mu V^{2/3})}$
2	800	2/3	0.80	$f_c = \frac{C_2}{29}\frac{\sqrt{(\mu V^{2/3})}}{P^{2/3}}$	$f_c P = \frac{C_2 P^{1/3}}{29}\sqrt{(\mu V^{2/3})}$
3	340	1/2	0.52	$f_c = \frac{C_2}{83}\sqrt{\left(\frac{\mu V^{2/3}}{P}\right)}$	$f_c P = \frac{C_2 P^{1/2}}{83}\sqrt{(\mu V^{2/3})}$
4	82	1/3	0.26	$f_c = \frac{C_2}{320}\frac{\sqrt{(\mu V^{2/3})}}{P^{1/3}}$	$f_c P = \frac{C_2 P^{2/3}}{320}\sqrt{(\mu V^{2/3})}$

of the previous deductions will be discussed from a more practical viewpoint.

Practical Applications

The value of factor n_o must be either known or assumed as a starting point, since this fixes the characteristic behavior of the bearing to be used. The derivation of equation (D7) shows that use of a close fit does not necessarily assure a high value of n_o; that is to say, of $K_o\sqrt{D}$. In practice, when the fit is usually made rather loose, n_o as taken from Table 7 may be expected to lie between ½ and 2/3.

The exponent n may now be appraised by means of equation (D10), after a suitable fit-constant K has been selected in accordance with Table 6. The exponent n varies with the journal diameter, and some idea of the intended size is taken for granted. Having interpolated n, its value may be substituted in equation (D13), from which the friction coefficient f_c can be predetermined when the velocity V and pressure P are known. The kind of lubricant generally required for any given service and unit-pressure load is rather definitely fixed by practice, and the corresponding absolute value of μ can be arrived at by consulting Table 2.

In some cases it is convenient to work on the basis of frictional drag fP, which represents the fluid-shear resistance between the rubbing surfaces as measured in pounds per square inch of projected journal area. Table 8 sets forth the different values of $K\sqrt{D}$, as determined from equation (D13), when $n_o = 0.64$ or $S_o = 0.8$.

The trend of the listed coefficients may be made evident by taking, as a concrete example, $V = 400$ and $\mu = ½$, and charting the results obtained by the Table-8 formulas, as in Fig. 16. In the case of the heavy-line reference-curve f_1, the pressure is assumed distributed over the entire projected journal area, but this can occur only when S and n both equal unity. When the f_c coefficients lie below this standard, such apparent reduction in friction is accounted for by a correspondingly higher P/S value, provided that this ratio is held within the critical-pressure limit P_c.

A curve for f_{exc}, as derived from my previously cited paper on High-Pressure-Bearing Research, for conditions in which the exponent n takes on a negative value, has been added to show the change in friction character after the film has been largely broken down.

Reasonably satisfactory results for moderately fast-running journals can be secured by assigning to the pressure exponent a preliminary value of $n = ½$. Also, since such journals may ordinarily be expected to run at a rubbing speed of from 250 to 750 ft. per min., the velocity factor can be fixed without serious error at a mean value of $V = 500$. An estimate for the corresponding averaged friction coefficient may be made by the use of the following equation, which is substantially the same as one appearing in a previous article of mine[14]:

$$f_p = \frac{C_2}{10}\sqrt{\left(\frac{\mu}{P}\right)} \qquad \text{(D18)}$$

The bearing constant or specific radiating capacity of a given journal for service of this kind will vary with the allowable temperature-rise. The rate of heat off-take is also dependent upon the massiveness of the bearing construction and the provision for artificial cooling. The following simplified relation assumes the natural or still-air radiating capacity of a bearing to be directly proportional to the projected journal area and to its temperature head, thus:

$$Ht_h = f_p PV = \frac{C_2}{10}\sqrt{(\mu P)} \qquad \text{(D19)}$$

The temperature rise t_h considered allowable is 40 to 60 deg. fahr. for heavy-duty bearings and 60 to 90 deg. fahr. for high-speed bearings.

When the conditions are such that the preliminary coefficient f_p is no longer applicable, a corrected f value is to be substituted in equation (D19).

Suitable allowances for natural or still-air radiation from stationary bearings to the atmosphere, as based upon $t_h =$ 80 deg. fahr. but with no allowance for the heat that may be removed by a copious supply of lubricant, are given in Table 9. For a normal still-air heat-off-take allowance, Ht_h is about 150 to 200 ft-lb. per min. The maxi-

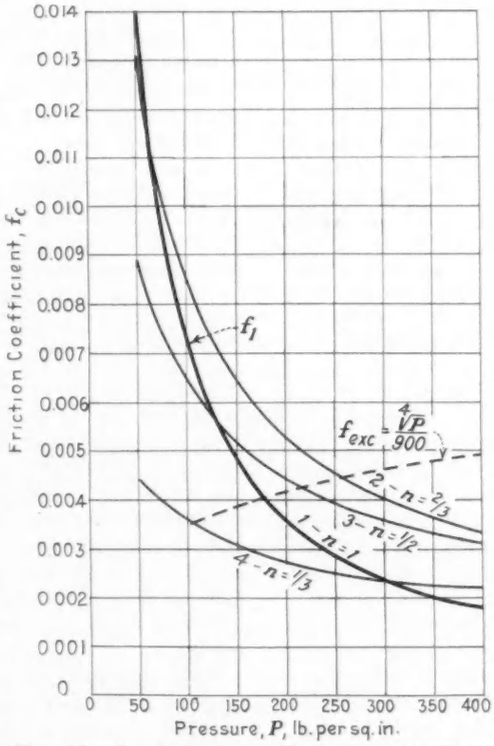


FIG. 16—COEFFICIENT OF FRICTION FROM FORMULAS IN TABLE 8

Based upon $V = 400$ and $\mu = ½$, for the Condition of $P < P_c$. Curves Are Numbered To Correspond with Line Numbers of Equations in Table 8

[14] See *Power*, Feb. 22, 1916, p. 251.

TABLE 9—APPROXIMATE RADIATING CAPACITY OF BEARINGS, IN FOOT-POUNDS PER SQUARE INCH OF PROJECTED JOURNAL AREA

H	Type of Bearing
2	Babbitted line-shaft-hanger bearings
3	Light-duty outboard or generator bearings without oil pockets
4	Small ring-oiled bearings with oil pockets
5	Heavy-duty outboard or generator bearings without oil pockets
6	Heavy-duty-engine main bearings without oil pockets

mum still-air value is about twice as much, and the allowance for crankpin bearings and the like should be 1½ to 2 times the still-air allowance.

While the specific bearing radiation H is taken as fixed, its actual value increases approximately with the square root of t_h. It is expedient, however, to reserve a certain leeway against overheating. The radiating capacity of any given bearing may be materially augmented by liquid cooling or forced ventilation, and this in turn correspondingly raises the allowable bearing constant.

The given still-air values for H apply to bearings that are in rooms and are not exposed to extrinsic air-draft; under such conditions, the resulting convection current is equivalent to a basic air-flow velocity of from 200 to 300 ft. per min. When freely exposed to normal outside atmospheric conditions, the equivalent expected draft may be taken at around 500 ft. per min. For bearings in motion, as in car journals, or that are otherwise subjected to forced draft, the resulting cooling effect H will be increased approximately as the square root of the ratio of the augmented air-flow velocity to the assumed velocity under still-air conditions.

Lubrication Efficiency

In the case of the basic f_1 reference equation, listed in line 1 of Table 8, the frictional drag is assumed to be independent of the pressure P. This condition can be satisfied only when n and n_0 both reach unity. The term P/S, as used in equation (D12), therefore applies solely to a bearing block in which the peripheral contact has been cut down to an arbitrary chord width S while the fit factor $K\sqrt{D}$ is maintained at its given critical value of 1250.

When the degree of contact is less than unity, because of a reduction in the fit constant K, the friction is determined by the coefficient f_c, and the resulting frictional drag will rise somewhat with an augmented unit pressure P. It is obvious that the accompanying exponent n will drop directly with the lowered value of K.

As compared with the reference standard f_1, the following relation serves as a measure of friction efficiency:

$$e_f = \frac{f_1}{f_c} = \frac{1}{(P/S)^{1-n}} \times \frac{1}{n^3} \qquad \text{(D20)}$$

[15] See *Automotive Industries*, Feb. 23, 1929, p. 290.

TABLE 10—APPROXIMATE DATA FOR CRANKSHAFT PROBLEM

Cylinder Size, in.	3½ x 4½ Main Bearing	Crankpin
Bearing diameter, in.	2¼	2
Bearing length, in.	2	1½
Rubbing velocity at 2500 r.p.m., ft. per min.	1500	1300
Mean pressure P from inertia, lb. per sq. in.	185	275

Certain of these values are plotted in Fig. 17 for the particular n to S relations listed in Table 8, showing a rather rapid e_f drop as the pressure increases. All of the curves cross the 100-per cent or reference line at about $P = 30$ lb. per sq. in., and thus indicate that low-pressure bearings may be expected to run to better advantage when the exponent n is kept below unity.

The reduced shear efficiency at lower n values apparently results from incipient contact between the metallic rubbing surfaces. When the fit is made loose enough

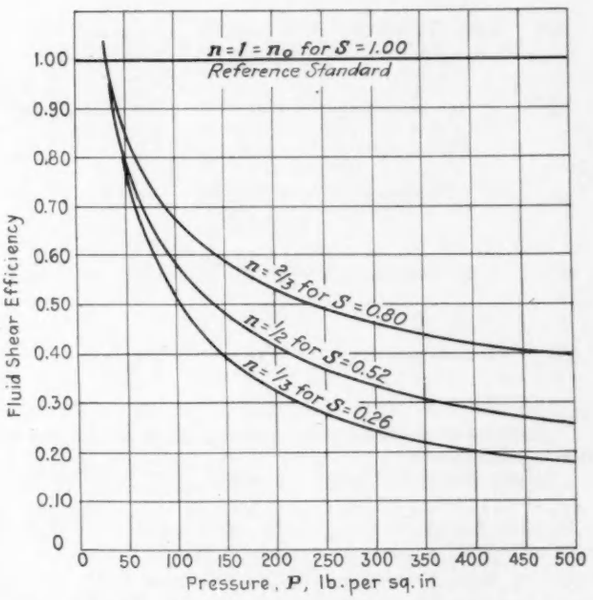


FIG. 17—FLUID-SHEAR EFFICIENCY
Plotted According to Equation (B20) for Various Values of n and S

to materially raise the pressure intensity P/S, there is, no doubt, a partial breaking down of the interposed oil-film that is not sufficient to throw the pressure exponent n into a negative value and thus carry the coefficient over into its intermediate-friction or f_{exc} range.

Automobile Crankshaft Bearings

Comparatively little research of importance has been directed to friction losses as found in automobile-crankshaft bearings. The fit of such bearings must be made close, because of rapid load reversal. Technical literature seems to be lacking in automobile-engine tests to determine the friction effect of the inertia pressures apart from piston friction. Lack of rigidity in the crankshaft mounting no doubt tends to raise the circumstance constant C_2 above that applicable to ordinary dead-load bearings, and this rather uncertain factor will be taken at an assumed value of $C_2 = 5/4$.

The foregoing deductions may be applied to a specific problem of a six-cylinder-automobile crankshaft. To this end, the tabulated data[15] for a considerable number of leading passenger-car engines were plotted, to find the prevailing crankshaft-bearing proportions, but this showed a rather wide divergence and lack of consistency in design constants. In the absence of uniformity of practice, a six-throw seven-bearing crankshaft of intermediate size was selected for check purposes. Data on the engine and crankshaft are listed in Table 10.

TABLE 11—BEARING DATA PERTAINING TO RAILWAY-

Reference Test No.	Experimenter[a]	Equation for Average Friction-Coefficient	Kind of Oil	Viscosity Equation[b]	Journal Size, In. Diam.	Length
B1	Beauchamp Tower	$f = 0.011\sqrt{\left(\frac{V\mu}{P}\right)}$	Rape	$\mu = \frac{21}{T - 50}$	4	6
B2	John Goodman	$f = 0.08\ \frac{\mu^{1/2}}{P^{1/4}}$	Rape	$\mu = \frac{20}{T - 50}$		
B3	A. M. Wellington	$f \approx \frac{6.5}{V^{1/3}\,P^{3/4}}$	Mineral		$3\frac{7}{16}$	7
B4	G. B. Karelitz	$f \approx 0.09\ \frac{\mu^{3/8}}{P^{1/8}}$	Heavy mineral	$\mu = \frac{20}{T - 90}$	5	12
B5	J. E. Denton	$f \approx 0.16\ \sqrt[8]{V}\sqrt{\frac{\mu}{P}}$	Paraffin	$\mu = \frac{10}{T - 50}$	3	6
B6	Composite	$f_r \approx \frac{1}{20}\ \sqrt[4]{V}\sqrt{\frac{\mu}{P}}$				

[a]References to publication or source of information are listed at the bottom of this page, arranged according to designations in the first column.

[b]These values have been computed or interpolated from given test data, as they are not recorded.

Inertia pressure due to the reciprocating parts will dominate the explosive pressure at high speeds. The problem may be simplified by assuming each separate main bearing to be individually loaded by only one of its adjacent power cylinders. The inertia pressure acting near the end of each stroke of a 3¼ x 4½-in. engine, running at 2500 r.p.m., can be taken at about 200 lb. per sq. in. of piston area; this in turn may be considered as producing an average reversing load on the crankpin and main bearing equal to the piston area multiplied by 100 lb. per sq. in.

Except for starting, it is unnecessary to consider the critical pressure P_c which, at the stated high speed, lies far above the stipulated mean pressure coming upon the bearings in question. The supply of oil needed to carry off the considerable excess of bearing heat is usually more than ample for lubrication.

In accordance with Tables 5 and 6, the fit constant K may be taken at 715; hence the $K\sqrt{D}$ value becomes about 1070 for the main bearing and about 1025 for the crankpin. Next, a suitable value must be assumed for n_o, which may be fixed at about 0.6, or $K_o\sqrt{D} = 750$, at $S_o = 0.78$. The fact that $K > K_o$ points to bearing friction that is likely to run rather high. To compensate for the closeness of fit, the friction coefficient f_c must first be determined from equation (D13), assuming the exponent $n = n_o = 0.6$, which is then to be multiplied by the ratio K/K_o. Taking $\mu = 1/5$, the resulting friction coefficient f_c is appraised at about 0.010 for the main bearing and 0.007 for the crankpin.

The friction losses for the seven main bearings and the six crankpins of this engine, running at 2500 r.p.m., can be calculated with the foregoing as a basis.

By multiplying the fPV bearing-losses, as taken in foot-pounds per square inch of projected area, by the number of bearings and their respective projected areas, a fair measure of the frictional horsepower can be arrived at thus:

Main bearing: $0.010 \times 185 \times 1500 \times 7 \times 2\frac{1}{2} \times 2 =$ 87,500 ft-lb. per min.

Crankpin: $0.007 \times 275 \times 1300 \times 6 \times 2 \times 1\frac{1}{2} =$ 45,000 ft-lb. per min.

Total crankshaft-bearing friction = 132,500 ft-lb. per min. ≈ 4 hp.

With 65 lb. per sq. in. brake m.e.p., the engine under consideration would deliver about 39 b.hp. at 2500 r.p.m.; therefore the crankshaft-bearing losses at high speed may amount to about 10 per cent of the rated engine power. The actual losses are probably somewhat lower, because the inertia pressures actually balance each other to a certain extent. Assuming such balancing effect to amount to about one-half, a rough check indicates the friction loss to be approximately 80 per cent of the stipulated 4 hp. At lower speeds and re-

LIST OF REFERENCES FOR EXPERIMENTS CITED IN TABLE 11, ARRANGED ACCORDING TO REFERENCE TEST NUMBERS

B1—Proceedings of the Institute of Mechanical Engineers, November, 1883, p. 632.

B2—Proceedings of the Institute of Civil Engineers, vol. 85, Part III, 1886, p. 376.

B3—Transactions of the American Society of Civil Engineers, vol. 13, 1884, p. 409.

B4—Transactions of the American Society of Mechanical Engineers, vol. 48, 1926, p. 1165.

B5—Transactions of the American Society of Mechanical Engineers, vol. 12, 1890, p. 405.

B6—Determined from Fig. 17 in this paper.

CAR OR PAD-LUBRICATED JOURNALS, WHEN $P < P_c$

Materials	Velocity Range, Ft. Per Min.		Pressure Range, Lb. per Sq. In.		Temperature Range, Deg. Fahr.		Absolute Viscosity-Range, Approximate		Remarks	Reference Test No.
	$V_{min.}$	$V_{max.}$	$P_{min.}$	$P_{max.}$	T_{min}	$T_{max.}$	$\mu_{min.}$	$\mu_{max.}$		
Steel on bronze	157	419	178	458	74	82	0.68[b]	0.88[b]	Chord of contact = 0.56 of pin diameter	B1
Steel on bronze	≈300	≈300	29[b]	225[b]	105	145	0.20[b]	0.36[b]	Given data unsatisfactory owing to incomplete report	B2
Steel on bronze	10	100	157	279	≈100	150	Estimated at ⅛ average		Pad initially flooded and bearing then allowed to heat up gradually. Given data found inconsistent	B3
Steel on bronze	174	720	6	31	120	175	0.23	0.83	Summer car-oil No. C here used. f taken independently of velocity.	B4
Steel on bronze	133	250	355	1660	85	110	0.17	0.30	Data difficult to coordinate	B5

duced inertia pressures, the friction losses would probably drop to about 6 per cent of the rated engine power.

It may be of interest to compare these findings with those reported by L. H. Pomeroy[16]. The series of tests that he reported was made upon a 3¼ x 5-in. six-cylinder engine and may be approximately represented by the following relation:

$$F = \frac{\sqrt{N}}{\sqrt[4]{(T - 90)}} \qquad \text{(D21)}$$

It will be seen that the considered losses increase as the square root of N but only as the fourth root of the viscosity component $T - 90$. A large and probably major part of such internal friction is due to the reciprocation of the piston, since its rubbing surfaces must be started anew with each stroke under unfavorable lubricating conditions.

The estimated crankshaft-friction loss of 4 hp. at 2500 r.p.m. represents a loss of about 10 per cent of 65, or 6.5 lb. per sq. in. of piston area. Assuming the engine to be running at 165 deg. fahr., the numerical value of F, as derived from equation (D21), is about 17 lb. From this it may be inferred that less than three-eighths of the internal engine friction is to be ascribed to crankshaft-bearing losses.

The minimum f_1 reference value, as determined by equation (D12), is only about 0.4 of the estimated crankpin and main-bearing coefficients; hence it seems likely that further reduction in crankshaft-friction losses will ultimately be brought about. In this connection, attention is directed to extensive research relating to the most suitable composition for bronze bearing shells[17].

Improved provision for lubricating car journals offers another fruitful field for research. Lubrication of these journals has received scant experimental attention, although they are so extensively used. Table 11 presents some of the more noteworthy tests that have found their way into regular channels of information. These data are plotted in Fig. 18, but the results are rather erratic and fail to show satisfactory agreement. A heavy-line composite curve, designated as f_r, has been added to represent present practice as nearly as it could be determined. A dotted curve for f_j is also shown, derived from the composite equation (A22) for the coefficient of friction, which lies very much lower than the f_r curve. This indicates that pad-lubricated car-journals run at far lower efficiency than do flooded journals such as most of those considered in Table 1.

Since the application of roller-bearings to railway-car journals is now assuming importance, two additional reference curves are presented. The dotted curve A in Fig. 18 has been deduced to represent the Stribeck roller-bearing tests[18] showing the higher efficiency, according to the equation

$$f_{s1} = \frac{0.055}{\sqrt{P_1}} \qquad \text{(D22)}$$

Another series of Stribeck tests shows a friction loss more than twice as great as that in equation (D22), and the corresponding friction coefficient is probably about $3/2\, f_s$ for the roller-bearing proportions prevailing in railway-journal practice. Curve B is based upon the roller-bearing findings of Thomas and others[19], which are of a somewhat different character:

$$f_{s2} = \frac{1}{180} \cdot \frac{\mu^{1/3} V^{1/3}}{\sqrt[4]{P_1}} \qquad \text{(D23)}$$

At pressures below $P = 100$, both roller-bearing tests show a friction coefficient that lies below the given f_j value; while, at higher unit pressure, the roller-bearing coefficients rise above those of the film-carried journal.

[16] See THE JOURNAL, March, 1924, p. 307; and TRANSACTIONS, vol. 19, 1924, part I, p. 227.

[17] See *Bureau of Standards Research Journal*, June 19, 1929, p. 1017.

[18] See *Zeitschrift des Vereines Deutscher Ingenieure*, Sept. 27, 1902, p. 1463.

[19] See Transactions of the American Society of Mechanical Engineers, vol. 35, 1913, p. 593.

The given f_s formulas merely represent a rough approximation for purposes of comparison. The friction coefficient obtained with roller-bearings actually depends upon a number of other factors, such as roller diameter and the pitch diameter of the cage, that have not been specifically taken into account. It appears, however, that the type superiority indicated by Fig. 18 can hardly be realized in railway-journal practice, partly because of the relatively high unit pressure that must be carried. Comparative tests were made upon a 40-ton box-car with the following results[20]: Starting tests show a drawbar pull of 38.6 lb. per ton of load for the plain bearing, as against only 4.4 lb. for the roller-bearing-equipped car. The power saving during a running test of some 28 miles at an average speed of 25 m.p.h. was estimated at 17.4 per cent.

Analysis of the f_r curve of Fig. 18 is difficult, because of insufficient data. The relatively high friction represented by curves $B3$ and $B5$ evidently results from a greater circumstance constant C_2.

Plain bearings of the car-journal type are snugly fitted or soon become worn to a close fit[21]. The standard journals are made in sizes ranging from 4¼ x 8 in. to 6 x 11 in. and are commonly loaded with 300 to 350 lb. per sq. in. of projected area[22], or an average of about 325 lb.

In the absence of an actual determination, the fit factor will here be taken at a limiting value of about $K\sqrt{D} = 1250$. The n coefficient should therefore lie close to unity; but, owing to a low n_0 value, the actual figure according to equation (B6) is only one-half of that magnitude. Using the equation listed in line 3 of Table 8, the friction coefficient, as determined for a unit pressure of $P = 325$ and under the μ and V conditions of Fig. 18, works out to about $f_c = 0.0031$. When this figure is multiplied by the stipulated K/K_0 ratio, to allow for abnormal closeness of fit, the coefficient is raised to about 0.005 at $C_2 = 1$. The corresponding plotted value by the composite equation (B6) is about 0.0081, thus indicating a circumstance value of $C_2 = 1.7$ for the best of the railway-journal tests, and C_2 is about 2.5 in the case of test B5. These relatively high C_2 values undoubtedly result from the restricted quantity of lubricant supplied to the journal by the pad type of feed.

According to the Hütte Handbook[23], the viscosity of the car-journal oil used in German railway practice during the summer season has the average characteristics of $\mu = 50/(T-55)$.

For winter oils, the C_3 constant is only about 6/10 as high. The summer oil at 180 deg. fahr. shows $\mu = 0.4$, and the same viscosity is reached at about 120 deg. fahr. for the winter oil. On the basis of a 5 x 9-in. journal, a rubbing velocity of 400 ft. per min. is set up when running at a train speed of about 30 m.p.h. Taking the mean viscosity at $\mu = 0.4$, an accompanying friction coefficient of $f_r = 0.008$ to 0.012 apparently represents expected railway-journal practice when $C_2 = 1.6$ to 2.4. Hütte[24] gives an f value of 0.009 to 0.010 for pad-fed rape-oil lubrication under a running pressure of $P = 275$ lb. per sq. in.

[20] See report by Timken Roller Bearing Co., May 23, 1925.

[21] See Proceedings of the Institute of Civil Engineers, vol. 226, 1928, part II, p. 284.

[22] See American Locomotive Co., Handbook, 1917, p. 103.

[23] Hütte, des Ingenieurs Taschenbuch, vol. 1, 1920 edition, p. 733.

[24] Hütte, des Ingenieurs Taschenbuch, vol. 1, 1902 edition, p. 214.

Concluding Remarks

Lack of efficient lubrication of drives is responsible, in the aggregate, for colossal motive-power losses. Continued research efforts along present lines may be expected to lead to a substantial reduction in the avoidable portion of such industrial waste.

This paper is believed to prove the need for proper coordination of this laboriously acquired test experience, to bring order out of chaos, and of putting these experimental findings into a form that will render them most helpful to the designing engineer. The promiscuous listing of heterogeneous test data generally found in current textbooks and handbooks is of little real value unless by chance one such test presents running conditions nearly identical with those sought.

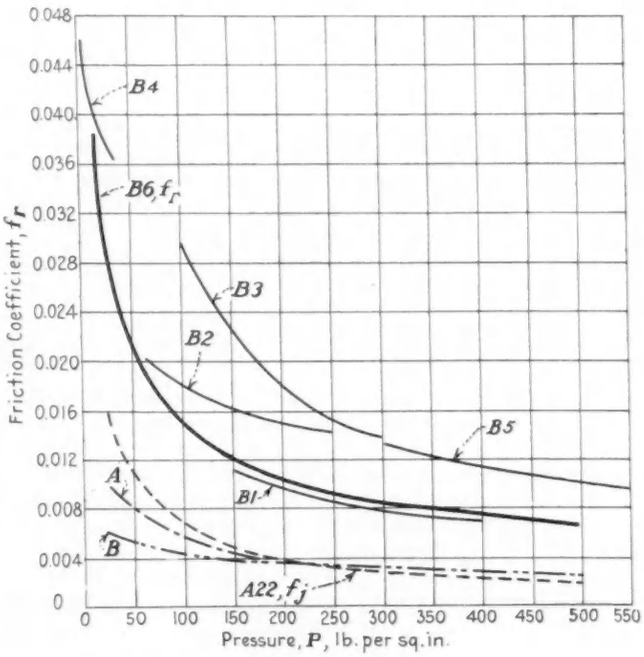


FIG. 18—FRICTION IN RAILWAY-CAR JOURNALS

Plotted from Equations in Table 11 for $V = 300$ Ft. per Min. and $\mu = ½$. The Heavy-Line Curve Is from the Composite Equation, and Gives What Are Considered Normal Values of the Coefficient of Friction for Railway-Car Practice. Composite Curve $A22$ and Curves A and B for Roller-Bearings Are Added for Reference

In time, the theory underlying friction will undoubtedly be placed upon a satisfactory basis; but the present fact-finding endeavor reveals the considerable difficulties encountered in a comprehensive study of this rather obscure subject. Further experimental investigation is needed before it will be possible to establish friction behavior, in its more involved forms, with certainty.

A Production Machine-Tool of Elastic Design

By J. B. ARMITAGE[1]

PRODUCTION MEETING PAPER — *Illustrated with* PHOTOGRAPHS

THIS paper was presented in the Production Forum Session to illustrate the adaptation of a standard machine-tool to automobile manufacture, as worked out by a milling-machine manufacturer. Beginning with a milling-machine of relatively simple design, special heads are substituted or added according to the requirements of the job. Adding a duplicate head at the front makes the machine duplex, and one or more adjustable heads for vertical spindles can be supported between the two main heads.

Discussers explained the ultimate advantages of using machine-tools of standard or special design. Many of the special tools are made up largely of standardized units and have high salvage value.

DESIGN changes in automobiles, as in all other mechanical products, are coming faster and faster. The span of time between abstract research and practical use is constantly decreasing. No sooner do we see the last of the planetary transmissions and begin to think that the three-speed sliding gearshift is practically standard than the trend veers to four-speed transmissions. Coming through the fog we see the front-wheel drive, the semi-Diesel-type engine and other major changes in design.

We know that designs must change, and it devolves upon the machine-tool men to make it possible for these changes to be put into production speedily and at the minimum cost. Our part is to furnish machines to produce this year's model and that will not be just so much scrap next year.

If production men want the maximum salvage value, they will use standard machine-tools entirely, but this is not practicable for quantity production; it generally entails too much handling of the work, too high a labor cost, and too much wasted floor space. Special machines performing several operations at one time are absolutely necessary; but they very often are nothing but scrap when the product is changed, so we see the economic reason for the present trend in the design of special machine-tools. This is to build special machines from standard machines and various units of standard machines as far as possible, keeping the special parts of the machines to the minimum.

In this way are secured low cost of the product, low first cost of the machine, and maximum salvage value when a change in the product makes obsolete the operation for which the machine was designed. When this time arrives, the machine can be returned to the builder, obsolete units discarded or changed, new units added if needed to suit the requirements of the changed part, and the entire machine overhauled and returned to the automobile builder, ready to take its place in the production line for the new car model.

This scheme is not a dream. Two years ago we presented to the trade the first models of machines built with this idea specifically in mind. Some of these machines have already been remodeled and are again at work on the new car models. We expect that in the future this salvaging and rehabilitating of single-purpose machines built on the unit plan will be a considerable part of the business of many progressive machine-tool plants.

A Simple Theme with Variations

How this interchangeable-unit idea is applied to the milling-machines we manufacture is shown in the accompanying illustrations. Fig. 1 represents the basic machine with a single horizontal spindle. We call this the Simplex machine. It has at the top of the head a pick-off change-speed box that is interchangeable with an optional quick-change-speed box.

The same basic machine provided with another type

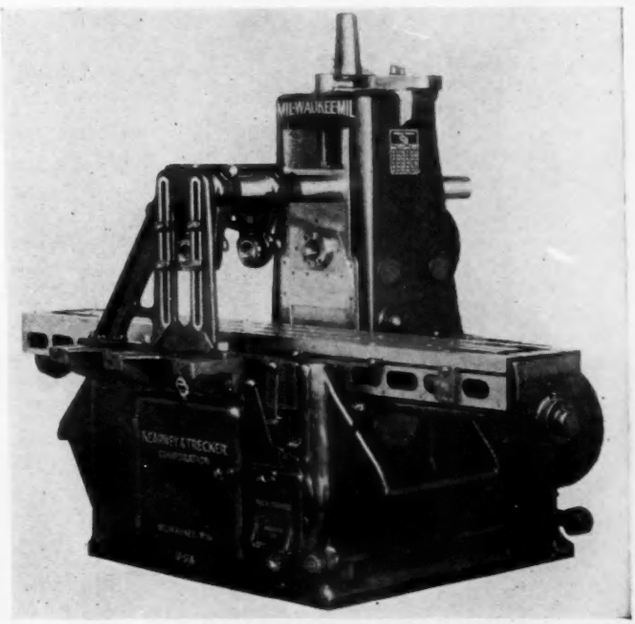


FIG. 1—KEARNEY & TRECKER SIMPLEX MILLING-MACHINE

[1] M.S.A.E.—Chief mechanical engineer, Kearney & Trecker Corp., Milwaukee.

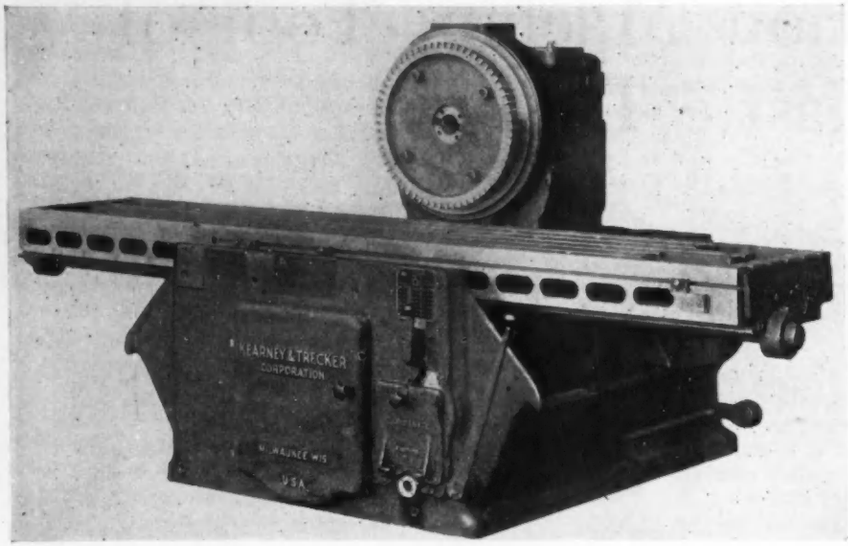


FIG. 2—SIMPLEX MILLING-MACHINE WITH SPECIAL HEAD AND LARGE FACE-MILL

of head which carries a 24-in. face-milling cutter is shown in Fig. 2. Heads of various types carrying one or more vertical spindles have also been provided. One of them is shown in Fig. 3. This is a complex unit having three spindles, one carrying a horizontal arbor provided with a special support.

Simultaneously straddle-milling both ends of a connecting-rod is the job of the machine shown in Fig. 4. This is a simplex machine provided with two spindle heads and four horizontal spindles, with an independent horizontal adjustment for each spindle. Two opposed spindles are included in the attachment shown in Fig. 5, also a vertical spindle. This machine mills the two sides and top of a clutch-housing.

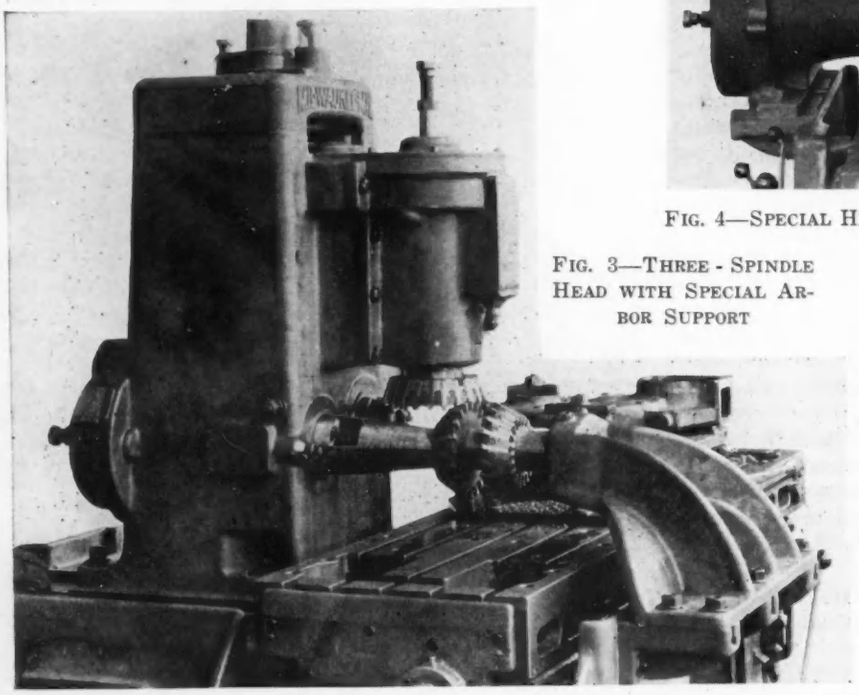


FIG. 3—THREE-SPINDLE HEAD WITH SPECIAL ARBOR SUPPORT

Duplex and Triplex Models

The standard Duplex milling-machine is similar to the Simplex, with the addition of an extension at the front of the bed which carries an additional spindle head. Fig. 6 shows such a machine provided with "double-decker" heads. Each of the four spindles has independent vertical and horizontal adjustment and can have any one of three different speed-ratios. Duplex machines sometimes have air-operated receding spindles to allow the work to be inserted in the fixtures. Fig. 7 shows two views of a more complicated arrangement, including four horizontal spindles, one vertical spindle and one angular spindle.

Another variation available is known as the Triplex machine, which

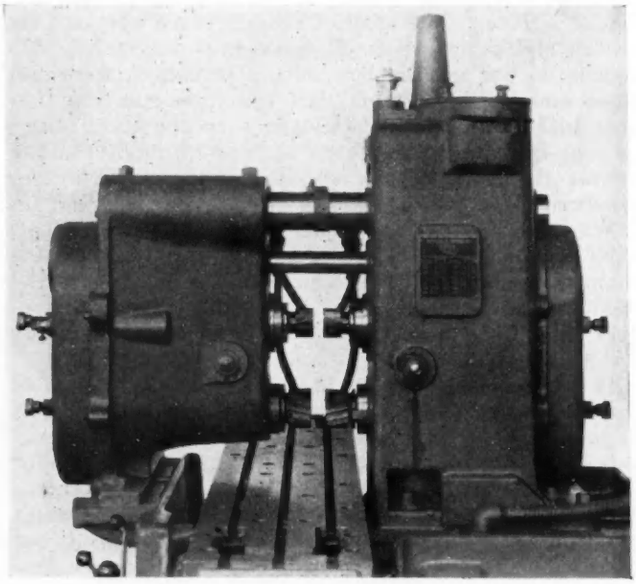
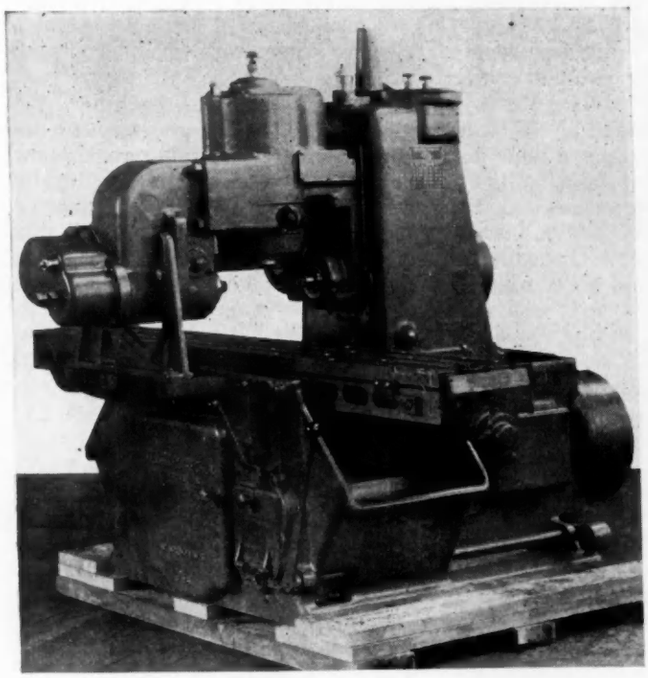


FIG. 4—SPECIAL HEADS FOR STRADDLE-MILLING CONNECTING-RODS

is made by adding, between tops of the heads and the speed-change boxes of the Duplex machine, a pair of yokes with carrier bars that carry one or two vertical spindle-heads. One of the Triplex machines is shown in Fig. 8, at work on a transmission case. The rear head carries a special spindle-block having two horizontal spindles. In addition, one vertical spindle and one horizontal spindle are provided on the front end. The transmission case shown extends downward through its holding fixture to within ½ in. of the top of the table.



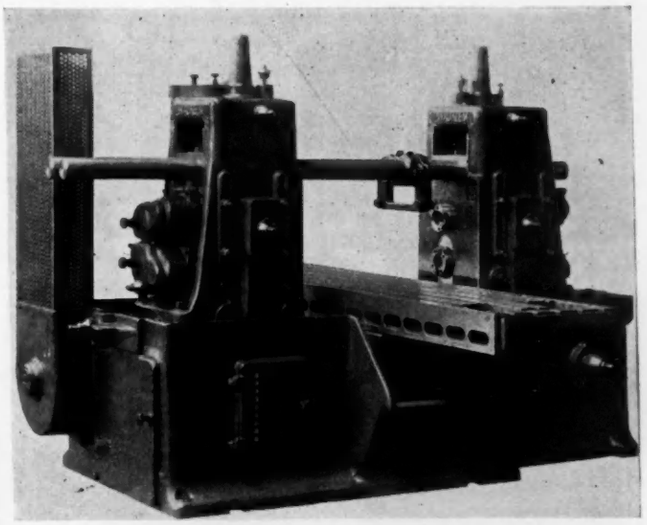


FIG. 5—SIMPLEX MACHINE FOR MILLING TWO SIDES AND TOP OF A CLUTCH HOUSING (LEFT)

FIG. 6—DUPLEX MILLING-MACHINE HAVING FOUR SPINDLES WITH INDEPENDENT HORIZONTAL AND VERTICAL ADJUSTMENTS AND SPEEDS (ABOVE)

THE DISCUSSION

E. P. BLANCHARD[2]:—Few automotive plants can afford special machinery that is intended for one job alone. The work can better be done by some machine-tool of more or less general application. Such a machine can be adapted to a new part when the manufacturing schedule changes. If the machine is not suitable to make the same part in a new design, it can be used for some other part. One user of chucking machines found that the expense was not excessive for re-tooling and adapting machines to a new model after approximately 15 years on one model.

A number of automatic machines now on the market can be adapted in this way, not only for changes in model but every few hours, to do a variety of work. Prof. F. E. Raymond's paper on Manufacturing Control through Economic Size of Production Lots[3] helps to show when such a machine becomes a paying investment on a group of jobs that are within its scope.

A machine that has a definite field of application is decidedly economical. Even if a change in production should make such a machine useless in the program of its owner, its resale value is much greater than that of a less adaptable machine.

WINFIELD S. HUSON[4]:—Both standard machines and special-purpose machines have their place. In the past, products were designed to be within the capacity of

[2] M.S.A.E.—Assistant sales manager, The Bullard Co., Bridgeport, Conn.

[3] See S.A.E. JOURNAL, November, 1929, p. 453.

[4] Mechanical engineer, Carboloy Co., Inc., New York City.

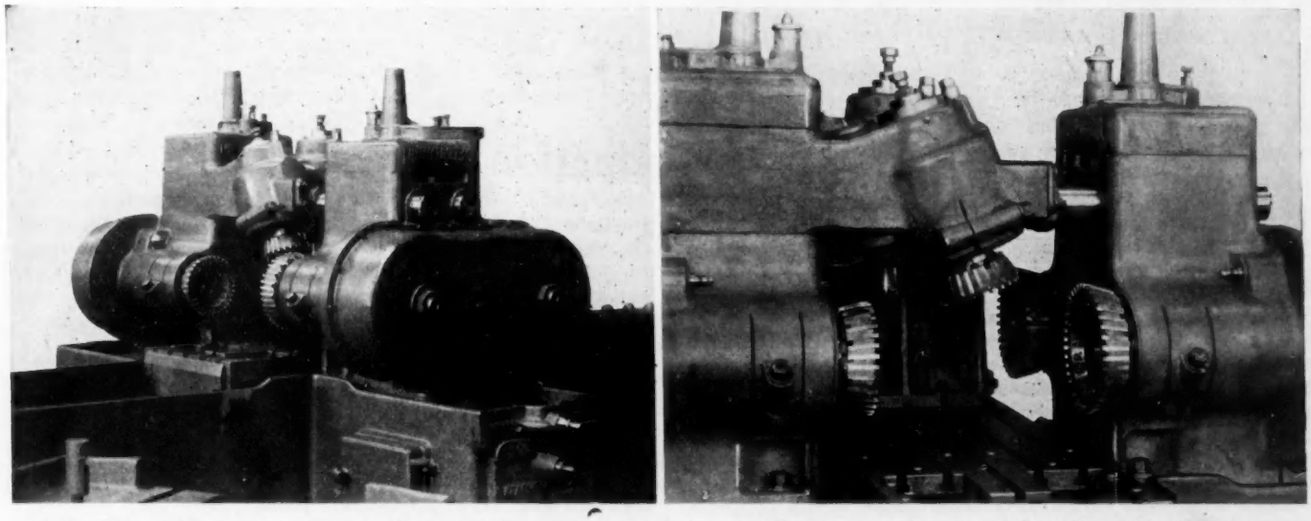


FIG. 7—TWO VIEWS OF DUPLEX MACHINE CARRYING FOUR OPPOSED HORIZONTAL SPINDLES, ONE VERTICAL SPINDLE AND ONE INCLINED SPINDLE

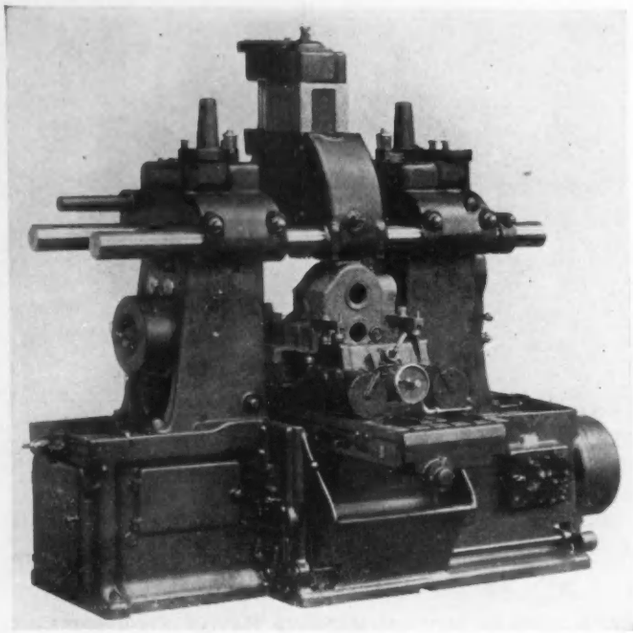


FIG. 8—TRIPLEX MILLING-MACHINE WORKING ON A HIGH TRANSMISSION-CASE

machine-tools. Now, in mass production, this limitation is removed and machine-tools are built to meet the demands of the product.

Special-purpose machines may seem to be something new, because the automobile industry is new; but thousands of special-purpose machines have been made that have never received publicity because they have been confined to some specific plant. Without doubt, the human equation is being eliminated more and more in the operation of machine-tools by the mechanism of the machines themselves. This is not pushing workmen out of jobs, because it leads to a higher economic condition with yet more machines and more leisure to enjoy the luxuries that high wages provide.

It is a question if we do not sometimes try to put into series operations that hamper one another. Perhaps we could secure larger production in some cases from individual tools if fewer of them were grouped around a common center.

JOHN MCGEORGE[5]:—What is a special machine? We have levers, wheels and pulleys. When an engineer designs a new machine, does he not simply make new arrangements of these well-known elements? Hydraulic transmission, for instance, is simply an arrangement of cylinders, plungers and pipes. Why should we call it a new transmission when it is simply an adaptation of old parts rearranged to fit new conditions?

Parts are made by boring, planing and cutting in various ways. We use only the old motions arranged in new ways; why cannot the designer think a little further and use standard tools to do this cutting, making the new arrangements to suit the standard tools?

Salvage Value of Special Machines

D. M. MILLHOLLAND[6]:—Sometimes what looks like a special machine is in reality a standard machine in that its basic construction is standard. It may be made of standard units mounted on various bases. The location of these units may be changed to accommodate the part being made. Such machines are not intended to take the place of conventional machine-tools, and they have no place in a job shop or in a plant in which frequent changes in production are made. They are designed primarily for a season's run on a part or a series of parts that require similar operations and can be accommodated by only slight changes in fixtures.

Such machines are called special-purpose machines, and most of them are so new that they are being used still in their original form. However, they can be taken apart and the units rearranged just as a boy takes apart the elements of a street-car that he has made from a mechanical toy and reassembles them into a motor-truck.

Applied to drilling-machines, with which I am most familiar, it may be necessary to discard fixtures and multiple heads having fixed center-distances. The fixtures would have to be thrown away in any case; and a scheme could be worked out for salvaging the spindles and bearings of the multiple heads, merely throwing away the heads. The salvage value is therefore very high.

[5] Consulting engineer, Ernest McGeorge, Inc., Cleveland.

[6] Partner, Millholland Sales & Engineering Co., Indianapolis.

Gold Production

SINCE the discovery of America, that is, during the years 1493 to 1927, inclusive, world production of gold has only slightly exceeded a billion ounces, approximately 1,003,500,000 oz. being indicated by study of available records. Some doubtful quantities have been excluded from this total; also, some production occurred in countries for which no records are available previous to the nineteenth century, as in the case of China and Japan. It is thought that all such unestimated production might reasonably be from 5,000,000 to 10,000,000 oz., representing, as a maximum, no more than 1 per cent of the estimated total.

More than half (516,273,000 oz.) of the grand total for the last 435 years was produced in the first 27 years of the present century; and, as this astounding increase represents only the culmination of abundant production that commenced with the discovery of gold in California, the world production of the last 27 years might be studied together with that of the previous 50 years. The total for 1851 to 1900 (336,000,000 oz.) and for 1901 to 1927 (516,000,000 oz.), covering 77 years in all, is found to make a grand total of 852,000,000 oz., as against 151,000,000 oz. produced in the 358 years from 1493 to 1850. If the entire world production since the discovery of America were cast into a cube, its edges would measure 38.5 ft.—Scott Turner, Director, U. S. Bureau of Mines.

Maintenance of Air-Cooled Engines in Commercial Operation

By Edward Hubbard[1] and L. S. Hobbs[2]

Chicago Aeronautic Meeting Paper

MAJOR principles of maintaining the engines of a fleet of airplanes are indicated, as determined for the passenger and mail service operating from Chicago to and along the Pacific Coast.

Fuel selection is one of the important decisions, and it is more difficult to form a basis for judgment of fuels than of lubricating oils. The cost must be weighed against possible damage from inferior fuel.

Temperature controls are emphasized for their effect on economy. They should be placed in the cockpit because of altitude and temperature changes during a single flight. Provision for oil-cooling, in which the regulation is effected with the control of engine temperature, is described. Present conditions as to ice formation in the carbureter are outlined, and the practical remedy suggested.

Boeing operations are under more nearly full-load conditions than is recommended. Under these circumstances, fairly frequent major overhauls have been found most economical.

COMMERCIAL operation of airplanes on a large scale began in this Country at the time when the single-row radial air-cooled engine had just proved itself; and it seems probable that this type of engine will predominate in commercial aviation, at least for several years to come. The Boeing Air Transport was the first commercial company to begin operations with modern airplanes built for the purpose, powered by radial air-cooled engines of the latest type and operated by personnel of exceptional experience.

Mail had previously been carried by Government-operated airplanes having 400-hp. water-cooled Liberty engines. The 400-hp. Wasp engines first used in these operations saved about 500 lb. in powerplant weight, this saving being available for extra pay-load. The new airplanes carried practically twice as much pay-load as the older ones, over the same lines and at the same or a little higher speed.

The Boeing operations are fairly representative in conditions of climate and terrain. The altitude varies from sea level to 14,000 ft., the temperature from the heat of the western desert and the midwestern prairies to the cold of high altitude in winter over Wyoming and Utah, and the humidity from the dryness of the desert to the extreme moisture of the Lake region around Chicago during certain seasons. Fogs are common over the entire line; snowstorms and blizzards are frequent over most of it during the winter.

One important factor affecting all data from this line should be considered in any analysis of the results or any attempt to utilize them for other operations. The engines are operating at an average power-output much greater than that generally considered desirable or economical from a maintenance standpoint, because of the required average speed and unpredictable increases in mail load. A careful record, kept since the beginning of operations, shows the cruising horsepower demand to be at least 85 per cent of the rated power on the average, and more than 90 per cent for considerable periods. Unfortunately, consolidated figures are not available on operations of comparative magnitude in which a normal output of 60 to 65 per cent of rated power is utilized; there are only isolated examples.

The Boeing Air Transport uses Pratt & Whitney air-cooled Wasp and Hornet engines exclusively. The Wasp is rated at 410 hp. and the Hornet at 500 hp. for commercial operation.

The maintenance of any engine in commercial operations is an economic problem, and its solution lies in the correct determination of the following points:

(1) Fuel to be used
(2) Lubricating oil to be used
(3) Engine operating conditions that are controllable by pilot or equipment
(4) Special equipment to meet special conditions
(5) Routine checks and adjustments
(6) Operating time before overhaul
(7) Kind of overhaul
(8) Allowable limits on wearing parts of the engine before replacement
(9) Replacement parts to be carried in stock
(10) The stage in the life of an engine at which it should be scrapped rather than rebuilt with new parts

Fuel Selection Is a Major Problem

Determining the fuel to be used is probably one of the operator's most vexing problems. A difference in cost of only a few cents per gallon becomes a sizable yearly total when the consumption is 1000 gal. or more per day. To be weighed against this is the possible cost of damage from the use of an inferior fuel, and no great imagination is required to visualize the cost of a forced landing and resultant crash brought on by a burned piston. In passing, we note that neither the Boeing Air Transport operations nor any other operation that has come under our observation has ever had this occur when operating on a California-base gasoline.

The fuel situation is aggravated by a serious lack of knowledge on the part of both the engine manufacturer and the refiner. The former does not know what actually occurs in the cylinder, and the latter has no real yardstick for measuring the quality of his fuel.

Fuel specifications for the routine standards such as distillation curve, gravity and freedom from acid are

[1] Vice-president in charge of operations, Boeing Air Transport Inc., Salt Lake City (Deceased).

[2] M.S.A.E.—Research engineer, Pratt & Whitney Aircraft Co., Hartford, Conn.

worked out well on paper, but there is no common standard for determining antiknock value. Even if there were, the proof is not definite that detonation is the sole cause of trouble. Under these conditions, the operator has almost no alternative; his only safe course is to use the fuel that experience has shown to be the best, regardless of cost. The extra cost for the best fuel usually is not prohibitive and is more than offset by freedom from certain kinds of possible failure that might result from the use of unsuitable fuel.

Choice of Oils Is Less Difficult

The choice of suitable lubricating oil is much more easily made than that of the fuel. More is known about oil, it can be well specified on paper, and ordinarily intelligent observation will make it possible in most cases to rectify a mistake before serious damage is done. Many good oils are available. There is much competition among the different refiners, and they are constantly and earnestly attempting to better their products. As in the case of fuel, the best oil is the cheapest in over-all operating cost. The best does not necessarily carry the highest cost per gallon.

The oil to be used should meet the specifications for a good oil, with particular attention to the viscosity and freedom from carbon. It should have high flash and fire-points, and should have a low pour-test for winter and cold-climate operations. It seems desirable, from our experience, to use a higher-viscosity oil in air-cooled engines in summer than in winter; about 120 sec. Saybolt at 210 deg. fahr. in summer and 100 sec. in winter are suitable. The condition of the first engine to reach 300 hr. will serve as a good indication of the quality of the oil, and 400 or 500 hr. will definitely determine it.

Carbon deposit assumes its greatest importance when engines are so loaded that they can be operated for long periods without overhaul. In the case given in detail later, in which an engine was operated for 525 hr. without overhaul of any kind, the condition of all parts indicated that the oil used would normally be considered very good; however, the carbon deposit had so filled the oil passages that another 100 hr. or so of operation would probably have stopped them completely, with consequent serious damage.

Economy Demands Temperature Control

Suitable control of such operating conditions as maximum and minimum oil temperatures and oil pressures, use of the mixture control, warming-up speeds and periods, and general handling of the engine are rather well defined from the engine manufacturer's experience.

Our experience has proved conclusively that the control of general engine temperature is of the greatest importance, and is by far the most important problem confronting the operator today. The temperature at which the engine operates has a bearing on the cost of maintenance, aside from the impossibility of operating the engine under some conditions of extreme temperatures unless temperature control is provided. Engines running too cool will show excessive wear, because of both raw gasoline getting into the cylinders and distortion of the metal. Engines running too warm will show considerable wear on all parts, due to inadequate lubrication from oil that has lost its quality because of excessive heat, as well as a greater carbon deposit from higher oil consumption.

The control of engine temperature divides itself automatically into two parts: control of the engine proper, especially the cylinders, and control of the intake-air temperature. The control of cylinder temperature must be effected by a cowling arrangement, which has generally been left to the airplane designer and the operator to work out. This attitude was the result of a genuine lack of knowledge on the part of the engine builder as to just what the requirements were. With the knowledge gained by the experience of the operator, standard controllable cowling and exhaust-manifold forms are being developed for the engines, and rapid strides in the art of cowling are being made.

The cowling should conform somewhat to the shape of the fuselage, and the cooling of the engine must be done without too great a loss in efficiency of the plane. Both the Boeing Air Transport and the manufacturer of the airplane have done a considerable amount of experimental work in an effort to combine these requirements with those of proper cooling and control. The most satisfactory installation in use at present consists of a fairly large nose-cowling with adjustable shutters. The cowling comes up on the cylinders practically to the heads. A good operating-temperature can be maintained in the coldest weather when the shutters are completely closed, and the area available for air-flow when open is greater than the space between the cylinders. An ample area of louvers, facing to the rear, provides for the escape of the air from the cowling.

It has been necessary in some installations to use oil-coolers because of the high atmospheric temperatures. These have been combined with the cowling so that the control of one cares for the other. The oil-cooler is placed between the adjustable nose-cowling and the engine, is annular, and extends almost completely around the nose of the engine. When the shutters are open, the same air passes through the cooler and around the engine.

Cockpit Controls Are Required

Sufficient experience has now been gained to enable the engine manufacturer to advise the user definitely as to cooling requirements when conditions of operation are known. Our experience has conclusively proved that all controls must be operated from the cockpit, as the conditions encountered during one flight will sometimes require wide variation in adjustment. The arrangement just described is particularly good in that it gives a single control to the pilot, who is overburdened with duties aside from those of piloting.

The control of the inlet-temperatures of air at the carbureter and mixture at the engine is a difficult problem in itself, and it is aggravated by the large volume of air to be heated and the small space available for heating. The hot-spot method of heating the mixture after it leaves the carbureter is quite effective from the standpoint of correct distribution of the mixture to the cylinders, but it does not care for one of the most serious problems encountered by the operator, that of ice formation in the carbureter. Formerly, ice built up principally around the throttles and throttle shafts; now the direct heat transmission from the hot-spot to the carbureter prevents freezing at the throttles, but ice forms around the carbureter discharge-nozzles in the venturis.

The range of outside-air temperatures which tend

to produce ice when the relative humidity is much above zero is between 65 and 32 deg. fahr. Above this range, the temperature drop in the carbureter is not sufficient to cause freezing, and water vapor cannot exist in air much below it; consequently, ice trouble is more prevalent in the spring and fall.

It seems that formation of ice in the bottom half of the carbureter can be prevented only by application of a suitable amount of heat to the air before it enters the carbureter. Application of a limited amount of heat when the air temperature is below 32 deg. fahr. may cause ice formation when it otherwise would not occur. Control of the preheating should be possible from the cockpit.

Special equipment to meet some special condition must in almost every case be worked out by the operator in conjunction with the engine manufacturer. A good example is the condition of abrasive material entering the engine through the carbureter intake and causing rapid wearing of parts. Fortunately, the landing-fields along the line of the Boeing Air Transport are such that this trouble has never been encountered. When it is present, a suitable air-cleaner can probably be supplied by the engine manufacturer or an approved type secured from an accessory manufacturer It also appears probable that an oil-filter, such as is now used on motor-cars, would be of great benefit when operating under conditions of abrasive dust, as much of the undesirable material could be separated out of the oil instead of being recirculated.

Checking and Overhauling Routine

With experienced mechanics and proper policy, routine checking is simple. We have found that an airplane should be checked at the terminal after each flight, including inspection of all visible parts of the engine and all engine controls, instruments, and gas and oil lines, with a final "run-up" of the engine after the inspection.

Notations should be made on an inspection sheet, showing any work done or adjustments made. This inspection sheet is shown to the pilot next assigned to fly the plane, so that he may be familiar with the work done and know that that plane has been inspected. The inspection outlined is made at each terminus of the plane, which means, in our service, at the end of each 4 or 5 hr. of flying. No other inspection is made except at the general overhaul.

The decision as to the time of operation before overhauling and the kind of overhaul to be made requires experience under the general operating conditions encountered. The Boeing Air Transport obtained this information in the following manner:

> One engine was run for 150 hr., removed from the plane, and given a complete overhaul, consisting of taking down the entire engine and all accessories. All parts were cleaned and measured for wear. Gaskets were the only replacements necessary. A close check was kept on the man-hours required for this overhaul, including removing the engine from the original plane and installing it in another.
>
> A second engine, which had been operating for 150 hr. under the same conditions, was given a top overhaul without removing it from the plane. This consisted of removing the cylinders and pistons, cleaning, removing carbon, and grinding the valves.
>
> A third engine was run slightly more than 200 hr. before being removed from the plane, and was given a major overhaul. The man-hours required for this overhaul were slightly more than for the 150-hr. period.
>
> A fourth engine was run 300 hr. before overhauling, and the cost of overhaul was found to be greatly in excess of the cost of overhauling the engine that had run 200 hr., due largely to the greater number of replacement parts, which necessitated more man-hours.
>
> Next, the engine that had had a top overhaul at 150 hr. was given a major overhaul at the end of an additional 100 hr., making a total of 250 hr. One bad bearing was found in this engine, with indications that the bearing had been in bad condition for some time. The replacement parts, aside from the bearing, were of minor character.
>
> Then the engine which had been given a major overhaul at 150 hr. was given a second major overhaul at the end of an additional 150 hr. This engine was found to be in good condition, and the only replacements necessary were piston-rings. The man-hours required were only slightly more than for the first overhaul, just enough additional time to replace the rings.

Frequent Overhauls for Economy

The conclusion reached by the Boeing Air Transport was that, for greatest economy in its operations, the engines should be overhauled at periods of 150 hr. We have further supported this theory by overhauling engines four times at periods of 150 hr., and the total cost of overhaul, including replacement parts, did not equal, on the average, the cost of overhauling the engine that had only one overhaul at the end of a 300-hr. period.

This policy can easily be carried out, because engines of this type can so readily be dismantled for frequent inspection. The old adage, "A stitch in time saves nine," can be applied, which means in this case that, with inspection after every 150 hr. of flying, a bad piston-ring, a scored bearing, a broken gasket or any one of a number of minor difficulties can be caught and corrected before it can result in excessive wear or damage to some major part, with necessity for costly replacements. The policy is further strengthened by the aim of the Boeing company to do all that is possible to safeguard operations and maintain schedules, and it is looked upon as a safety measure.

The engines so far given as examples have been operated at what is regarded as an excessive power output, as previously noted. The only figures available on engines that have been operated at a normal output are for isolated instances. One engine, in the nose of a three-motored airplane, was operated for 525 hr. at an output of about 58 per cent of its rated horsepower. The cost of overhauling this engine was low, the only parts requiring replacement being piston-rings, oil-strainer and gaskets. Two other engines, operating in small cabin planes in regular airway service, under severe conditions of low atmospheric temperature, had 350 hr. of flying time. Their average power output in operation was about 64 per cent of the rated horsepower. The replacements required in these engines were piston-rings and one piston. Many other individual examples are available, all falling between these two.

Top overhauls are impracticable, according to the experience of the Boeing Air Transport. The man-hours required to give a top overhaul of the engine in the airplane are nearly as many as to give a major

overhaul in the shop, including the removal and reinstallation of the engine. Man-hours on the engine in the shop are much more efficient than when the engine is in a plane. Again, a top overhaul in a plane is not regarded as so safe a method of procedure, because of the lack of inspection of all parts of the engine, which is done as a matter of routine in a complete overhaul. In addition to all of this, airplanes are expensive equipment and must be maintained in service at all times for the most economical operation. Removing an engine and installing another allows this, while the top overhaul puts the plane out of service for the time required to make it.

A major overhaul should consist of taking down the entire engine, together with carbureters, magnetos, generators, ignition system, and all other accessories. The limits to be allowed for the wear of engine parts are well defined by the manufacturer of the engine; following them will be found conducive to economy. Replacement parts to be carried in stock are almost definitely determined after the first few major overhauls or mechanical failures in flight. We have found it necessary to carry spare parts to the extent of about 3 per cent of the major parts and 10 per cent of the minor parts. These parts represent about 5 per cent of the total value of our 50 engines.

None of our engines has yet reached the stage where it is considered necessary to entirely scrap it because of general fatigue failures, except in the case of one major part, which was found to have fatigued to the breaking limit after about 800 hr. of operation and was immediately strengthened by the engine manufacturer. Therefore, we are not in a position to give definite figures for the ultimate life of an engine. The engines show no higher percentage of mechanical failures as operating time accumulates. Some minor parts wear out and are replaced, and trouble is not expected until such parts as crankcases and connecting-rods begin to show signs of fatigue. Master-rod bearings are replaced at regular intervals, and crank-pins show no appreciable wear.

Operations Based on Experience

In conclusion, we wish to add the following observations, based on experience to date:

For most satisfactory operation, the type of ship and its equipment must be selected and planned with a full knowledge of the conditions and purpose of operation.

Frequent change of oil, or some suitable type of oil-cleaner is desirable.

Careful attention should be given to routine checks before and after each flight and at other stated periods.

The labor required for complete overhaul of a radial air-cooled engine is approximately 40 per cent of that of a water-cooled engine such as the Liberty; and the average cost of parts to be replaced is considerably less, notwithstanding the fact that overhauls are required less frequently.

It is economy to remove an engine from the airplane for overhauling, rather than to overhaul it while mounted in the plane; although the Wasp and Hornet engines can be overhauled without dismounting, if the occasion requires.

The ultimate life of these engines has not yet been determined, as the average time of operation of Wasp and Hornet engines on the Boeing line is only about 1000 hr.

Experience is teaching the economy of reserve power in commercial operation. In addition to lengthening the time between overhauls and reducing the rate of wear on moving parts, reserve power reduces the percentage of full power required for normal cruising operation and provides the extra power needed occasionally for conditions of emergency, thus adding to the safety of flight.

All Passengers Challenge Engine Builders

(*Concluded from p.* 40)

that no engine manufacturer can afford to ignore. It is his duty to pay very close attention to the service requirements of his engine, with the purpose of eliminating them whenever possible or of simplifying the service operations where they are necessary.

To recapitulate, the requirements of the air passenger seem to propose to the engine manufacturer the following courses of procedure:

(1) Use of reduction gears to improve propeller efficiency and to reduce propeller noise

(2) Development of in-line or V engines, liquid or air-cooled to reduce vibration, frontal area, and possibly manufacturing costs

(3) The utilization of exhaust mufflers

(4) Thorough investigation of the possibilities of burning heavy fuel

(5) Elimination of interference by ignition system with radio reception

(6) Constant attention to production problems in the manufacture of aircraft engines and to the reduction of cost in general

(7) Continuous effort to reduce the service maintenance required and to simplify performance of the service which cannot be eliminated.

The task is not easy, and the manufacturer labors on, with the assurance that, when these requirements are met, more will have been discovered.

Motorboat Fire Prevention

By H. G. Kamrath[1]

Motorboat Conference Paper

THE recent rapid rise in popularity of motorboating has emphasized the subject of safety. In past years, motorboating was a sport available almost exclusively to the wealthy who could not only afford the relatively high cost of the initial equipment, but who could secure the services of experts to operate and maintain this equipment. Times have changed, however, and mass production along with improvements in design and manufacture have placed pleasure boats within the reach of thousands of potential owner-operators. With the present acute congestion of State highways, especially on holidays, one must either take to the air or the water to derive any pleasure from going places. Airplanes are seriously handicapped because of the need for expert operation, but the motorboats of today are so foolproof that average motorists can and do operate them with no trouble whatsoever. Thousands of families that formerly took their week-end trips in the family car are now taking their week-end cruises in the family motorboat. That we discuss at this time one of the greatest hazards in connection with motorboating, the fire hazard, is therefore appropriate.

Fire in a boat, large or small, has always been a thing of dread. This is particularly true in the case of small boats where recourse to lifeboats is impossible and life belts must be used. In the latter event, exposure is often as great a hazard as the fire itself. Since the thought of such hazards will turn many persons away from buying motorboats, motorboat builders will benefit greatly by increasing safety factors of their products.

Causes of Fires

Possible causes of marine fires are as follows:

(1) Carelessness, due to unwise handling of matches and cigarettes. This is distinctly a human element and entirely beyond the control of the boat builders.

(2) Spontaneous combustion, caused by stowing oily rags and similar articles in lockers. This might also be classed as carelessness and is also an uncontrollable factor.

(3) Electrical, due to defective wiring causing short-circuits, defective high-tension insulation causing spark jumps, or defective electrical accessories such as a sparking generator or magneto. These items can be controlled by the manufacturer by careful selection of electrical equipment and its proper installation.

(4) Exhaust system, resulting from leaks caused by poorly designed joints or inadequate gasket material. This is a subject of design.

(5) Mechanical, caused by excessive friction in some mechanism or overheated exposed bearings due to oiling-system failure, inadequate bearing-area or misalignment. This is also a design item.

(6) Carbureter backfire or flames being blown out into the engine compartment during backfire.

[1] M.S.A.E.—Experimental engineer, A. C. Spark Plug Co., Flint, Mich.

This last cause and its cure form the subject of this paper. About two years ago the Underwriters' Laboratories became actively interested in the matter of arresting backfire flames as they emerged from the carbureter. Since automobile air-cleaners were attached directly to the carbureter air-sleeve, the Laboratories arranged to test commercial air-cleaners to determine their merits as flame arresters. The result was that none of the inertia or centrifugal-type cleaners so popular at that time passed the requirements. However, the surface-adhesion type of cleaner, consisting of a hollow cylinder of metal screen or mesh, successfully passed the tests. To convey an idea of the nature of these tests, I quote directly from the report, Automotive No. 1558.

Backfire Test

Method.—With the sample installed, the test engine was operated with retarded spark and a lean mixture to induce backfiring through the carbureter. This test was performed in a darkened room. Observations were made as the engine was continuously backfired to determine the effectiveness of the cleaner in arresting the backfire. A gasoline-soaked rag was placed near the samples to determine the range of the flame emitted from the cleaner.

Results.—The cleaner completely prevented propagation of the flame. No flames could be observed beyond the steel outer shell. The gasoline-soaked rag could not be ignited even though held within 1 in. of the shell.

Ignition Test

Description of Sample.—The cleaner employed in the foregoing test was used in these tests.

Method.—The cleaner gauze was filled with sawdust which had been saturated with a mixture of oil and gasoline. An attempt was made to ignite the material so as to start a fire in the cleaner by repeatedly backfiring the engine.

Results.—This could not be done. The material was gradually dislodged by the force of the backfire until, at the completion of the test, practically none remained on the gauze. The gasoline was vaporized soon after the engine started. Repeated and severe backfire flames emitted into the cleaner failed to ignite the sawdust.

Relation to Fire Hazards

This device will minimize the possibility of backfire flames being emitted into the engine compartment of an automobile.

Study of the design of the cleaner, together with a consideration of the Backfire Test, showed that the backfire flames are completely arrested in the cleaner and will not reach the engine compartment. Repeated and severe backfires failed to ignite flammable material held within 1 in. of the shell.

Inasmuch as a large number of automobile fires have been traced to carbureter backfire, it is thought that the attachment of a properly designed cleaner to the carbureter intake will materially assist in decreasing the number of fires.

Fire-Loss Data

A review of automobile fire-loss reports at the Underwriters' Laboratories showed that, of 403 cases over

(*Concluded on p. 97*)

Section Membership Values and Opportunities

By J. A. C. WARNER[1]

PITTSBURGH SECTION ADDRESS

WITHOUT direct participation in Section activities a man cannot take full advantage of what the Society has to offer. Our bodies recuperate comparatively rapidly from overwork and from too much exertion, but our spirits, our contacts, our association of ideas need to be revived and kept alive by contacts with persons who are interested in the same line of work in which we are interested. A man who is all wrapped up in himself usually makes a very small package. We certainly need fellowship, human contacts and the rubbing of elbows with our fellow men to keep us from becoming stale.

We often hear the remark made that it is impossible for a person to get out of an organization more than he puts into it. I maintain that a person can get a great deal more out of a local Section of the Society then he puts into it. Suppose, for example, that you, an average member, attend a Section meeting. What do you put into it? You put in the time, physical effort and energy to get to that meeting and get back home.

What do you get out of it? You get a number of very important things; first, a refreshing contact and thrill from the discussion, out of school, with men who are working in your own line. You gather a knowledge of facts gleaned by others which you could not gain in any other way. You get a broader view of engineering as a whole. You have an opportunity to engage effectively in the local promotion of worthwhile municipal and National undertakings that involve automotive work. Through Section contact you can join in the important and far-reaching activities of the National Society. You have an opportunity to enjoy a degree of relaxation, leisure and entertainment. In fact, the thrill of participating fully in the facilities and in what the S.A.E. is and has to offer cannot be derived unless you belong to a Section.

Second, although we do not try to represent the commercial value of the Section, if a person is so inclined, he can sit down and figure it up in dollars and cents. For example, I know a man who, after attending a Section meeting, told me privately, "I like to listen to the papers and I get a great deal from some of them, but my chief interest in attending S.A.E. Section meetings is to talk over the fence with some of the men who are interested in my problems." This man further told me that he got one idea at one of the Section meetings that meant considerable financially to him directly and that each year put $50,000 into his company's pocket. You won't get one of those ideas at every meeting, I will guarantee, but you can, if you will, gather a great deal more than you put into your local Section.

These advantages cannot be derived from reading the S.A.E. JOURNAL once a month. In my opinion, an S.A.E. member who is not a Section member is only half a member.

I want to congratulate the officers and also the members of the Pittsburgh Section for the fine attendance we have here tonight. It is a surprise to me, and I gather that it is a surprise to the management. It did not "just happen" that this event is such a success. A meeting like this requires considerable effort and thought on the part of the officers. Any group of men who desire to have a successful Section must be willing to work hard. It is not a job that can be performed sitting behind a desk and, with a wave of the arm, presto! it happens. A successful meeting never "just happens."

Section Membership Increase Needs Salesmanship

One item about your local condition that I think must be of particular interest to you is membership. Perhaps you remember the story of the census enumerator who asked the housewife how many children she had. She answered, "I have eight." He said, "For God's sake, don't have another, because the statistics say that every ninth baby born is an African Zulu." Now the Pittsburgh Section has 68 members, but don't be afraid to get the 69th for fear he will be an African Zulu. Your Membership Committee has a real job on its hands, but I feel that if you, who are not members of the S.A.E. and of the local Section, realize that the men who are at the helm are there to produce something that is worthwhile to you, you are not going to be backward about joining; you cannot afford to miss it. So, with an active Section here in Pittsburgh, I predict that you will soon have 10 times 68 members.

You who manage this Section have something very definite to sell. Why not get out and sell it? I imagine that is exactly what you are doing. Once in a while we have in the Society a Section that is in what one might call a comatose state, just about ready to breathe one deep gasp and pass out. That usually is a case in which the officers and members have not understood the selling angle of organization advantages. My idea of an engineer, and the idea which the Society has of an engineer, is not that he is necessarily one who pushes the pencil and works the slipstick. He is an engineer, but there are engineers of various kinds, including the sales engineer. We are prone to think of designers and production men as the only real engineers, but in reality

[1] M.S.A.E.—Assistant research engineer, Studebaker Corp. of America, South Bend, Ind.; Chairman S.A.E. Meetings Committee.

the sales engineers are just as much engineers as are the others. That applies not only to the automobile trade but to the aviation field as well.

A Survey of Membership Potentialities

A question always arises whether a given territory has adequate material for a real lively S.A.E. Section. This morning, to obtain a scientific fact, I went through a very unscientific book that is published in Pittsburgh. That book was the telephone directory. I was interested to see how much of the classified part of the directory was automotive. You will be surprised to know that the automobile and the aircraft fields take up 75 pages of that classified directory, which is five or six times as much as any other single industry in the Pittsburgh territory requires.

I gather that Pittsburgh supplies the world with raw materials and finished products in coal, iron, steel, aluminum, alloys, glass, chemicals, lubricants and so forth without end; that you construct, sell, operate and maintain automotive vehicles and equipment, including aircraft; in fact, your transportation interests include every known means of operation on land, on water and in the air. I understand that Pittsburgh is the center of the electrical-equipment industry, as you distribute and handle everything from the smallest incandescent lamp to the heaviest electric generator; that you are preeminent as regards radio and other means of communication; and you have several of the world's greatest educational and research institutions: The Mellon Institute of Industrial Research, the Carnegie Institute of Technology, the Pittsburgh University, the Bureau of Mines Experiment Station, and others. I should say that you have a priceless basis for an unsurpassed S.A.E. Section.

In closing, I should like to use the words that Coach Bob Waddell will use on Saturday when he sends his warriors from Carnegie Tech upon the field against Notre Dame. He is going to say to his men, I venture to predict, "You have the wherewithal, you have the physique, you have the stamina to win." And last he is going to say, "Go, go, go!"

Motorboat Fire Prevention

(*Continued from page* 95)

a period of three years, 50 were traced directly to carbureter backfires. For the year 1928 and the latter part of 1927, approximately 10 per cent were traced to carbureter backfire. Of the 1928 fires, the percentage is 18. In the 35 cases so far reported for 1929, 11, or 31 per cent, were traced to carbureter backfire. All of these figures were from one large insurance company only. The apparent increase in percentage of fires resulting from backfire probably is due to the fact that widespread adoption of safety factors such as the fuel-pump have reduced the number of fires from other causes. At the present rate for 1929, the total for the next three years will amount to only one-half that reported for the last three years. Considering the increase in registration, a very marked improvement is indicated. With the recent adoption of the flame-arresting type of cleaner by many manufacturers, future reports should be even more favorable.

Fire-loss reports for motorboats were not available but the Underwriters' Laboratories is now conducting an investigation of this particular phase. I believe, however, that the percentage of marine fires caused by backfires will be relatively high compared with automobile figures, because a large percentage of automobile fires are the result of collisions or of rolling over into the ditch.

Flame Arrester Resembles Miner's Safety-Lamp

As previously stated, the only type of flame arrester approved as standard by the Underwriters' Laboratories is the screen or mesh type, arranged either in the form of a hollow cylinder with the air passing radially to the inside and then axially into the carbureter or in the form of a porous cylinder through which the outgoing flame is absorbed by the metal mesh at such a rate that it is cooled below the point of ignition. Consequently, only a gas emerges from the arrester and this is at a temperature below the ignition point of any other gas or substance with which it may come in contact. Sir Humphry Davy discovered the basic principle of the flame arrester in 1815 when he invented the miner's lamp. His lamp consisted of an open flame surrounded by a cylinder of metal gauze.

In the type of flame arrester manufactured by the company I represent, the heat-absorbing material is copper. This relatively expensive metal is used because its thermal conductivity is nine times that of steel, making it particularly suitable as a heat absorbent. The fine wires, instead of being round, are flat to present large surface with the minimum weight. This feature assures very rapid absorption as well as a very rapid loss of heat, which is of prime importance when repeated backfires are encountered. The thin flat wires are quickly cooled between backfires by the air charge passing into the carbureter. A further feature of this type of arrester is that the flat wires are knitted in an open mesh with the flat surfaces at every conceivable angle. Thus the flame, in passing into the mesh, is thoroughly and uniformly diffused throughout the entire mass. A further feature of this type of flame arrester is the louvered outer casing that protects the carbureter from excessive water in exposed engine-compartments.

The attachment of flame arresters to carbureters is usually made by a clamp and the standard flame-arrester outlets conform to S.A.E. Recommended Practice for air-horn sizes. The experience we have had to date with marine carbureters indicates that marine engineers have specified a large number of special carbureter air-sleeves with odd diameters. If standard S.A.E. air-horn sizes are adhered to in future designs, making possible the installation of a standard flame-arrester, the result will be very much to every one's advantage. When all engines are finally equipped with flame arresters, one of the most treacherous hazards to boat users will have been eliminated.

Beryllium as a Possible Structural Metal

By Dr. C. B. Sawyer[1]

Cleveland Aeronautic Meeting Paper

BERYLLIUM is a metallic element somewhat resembling both magnesium and aluminum but differing also from each. It has a peculiar combination of properties to justify most serious engineering attention. Chief of its excellent properties are: density, very low, about the same as that of magnesium; strength, high, and comparable to that of steel; modulous of elasticity about the same as that of steel; coefficient of thermal expansion and melting point about the same as those of cast iron; also, resistance to corrosion greater than that of aluminum.

But, with all this array of excellent properties, beryllium as a structural metal still has a serious fault; it lacks ductility. Without doubt, sufficiently ductile alloys consisting almost entirely of beryllium can, in the future, be prepared, and certainly the metallurgical achievements for tungsten are always before us as inspiration. However, there is this fundamental difference between the two metals; tungsten has the cubic-crystal habit, or space lattice, which places it in the group of very ductile metals including gold, silver, copper, aluminum, and so on. Beryllium, on the other hand, has the hexagonal-crystal habit, or space lattice, in common with zinc, cadmium and magnesium, none of which possesses great ductility. So evidently the task of preparing ductile alloys of very high beryllium-content is not easy, though alloys of somewhat less beryllium-content do give much encouragement. For instance, light ductile alloys of beryllium and aluminum containing up to 60 per cent of beryllium are already known and are reported to have a tensile strength of 80,000 lb. per sq. in., or even higher. Such alloys as these have a density nearer to that of magnesium than of aluminum, and should have a wide field of application.

For some uses it is not necessary that beryllium should have even moderate ductility. In the unalloyed and rather brittle state, as known at present, it could probably find immediate employment for pistons and castings, in which, because of its low density, beryllium could replace cast iron with great advantage. However, even for such a simple process as casting, much knowledge concerning methods of handling is needed. A 250-page book[2] on beryllium has been published recently in which much valuable information about the production of beryllium is set out. It also contains very interesting and suggestive data concerning heavier alloys of lower beryllium contents.

With the field for beryllium rather well assured, it is of interest to consider general conditions determining its probable cost. Such estimates as follow are not intended to reflect an exact study, but, by and large, they do indicate what may reasonably be expected when the development period has been passed and a fairly large-scale production established.

Though at first beryllium was regarded as a rare element, it has in fact not turned out to be so. Indeed, its position as No. 4 in the periodic table places it midway among the first nine elements, every one of which is known to occur in large quantities; and, as interest in beryllium has increased, so have known deposits of the ore, beryl. Several sources are now offering beryl in carload lots at prices from $50 to $60 per ton. Other sources will certainly appear. A steady contract with a reliable firm for considerable quantities would doubtless produce beryl at $20 to $30 per ton. Assuming that only 75 per cent of the beryllium in the ore can be recovered as metal, the ore cost per pound of the metal would then lie between 37 and 56 cents.

The composition of beryl is such that it reminds one of clay with part of its aluminum oxide replaced by beryllium oxide. The Bureau of Mines has shown that clays containing about 34 per cent of aluminum oxide should be capable of profitable extraction for aluminum oxide. Keeping in mind that beryl contains about one-half as much aluminum oxide as does clay, it is evident that the recovery of aluminum oxide as a by-product in the extraction of beryllium oxide will go far toward carrying the cost of such extraction, provided the process is simple. The company with which I am connected, has made a sufficient study of beryl to feel sure that such a process will be developed. It seems fair to assume that when large quantities are employed it will be possible to extract the beryllium oxide of beryl at a cost so low as to be unimportant in the total cost of producing the metal.

Let us assume that the process to be used for the reduction of beryllium oxide to the metal is one of electrolysis of fused fluorides, or fused chlorides. From electrochemical calculation, it is at once evident that 1 lb. of beryllium will require twice as much current as 1 lb. of aluminum. The amount of power required is uncertain but may be 10 times as much. To be extremely liberal, the whole electrolysis may be between 10 and 20 times as costly for 1 lb. of beryllium as for 1 lb. of aluminum. At the figure of 10 cents per pound of aluminum, as given by Anderson, the electrolysis of 1 lb. of beryllium would cost between $1 and $2.

Combining cost figures for ore and electrolysis, it seems that, on a large scale and with well-developed processes, a production cost between $1.40 and $2.70 per lb. of beryllium for labor and materials can ultimately be realized. This is well within practical limits of use and, accordingly, the beryllium future should command keenest attention.

[1] Brush Laboratories Co., Cleveland.

[2] Wissenschaftliche Veröffentlichungen aus dem Siemens-Konzern. Beryllium-Arbeiten. Published by Julius Springer, Berlin.

Automotive Research

Riding-Qualities Research

Six Tests of Muscular and Nerve Fatigue Selected as Result of Dr. Moss's Work

STABILITY of the human body after a ride may prove to be the index for evaluating the riding-comfort of various cars, and thence for predicting the effects of the various types of spring suspension, shock-absorber and other comfort-giving components. This possibility is indicated by the most recent developments of the Society's riding-qualities research project being conducted by Dr. Fred A. Moss, head of the Department of Psychology of George Washington University, City of Washington.

Working on the assumption that comfort is the same as lack of fatigue, and using the human body as a measuring-stick, Dr. Moss devised a whole series of tests for measuring body changes brought about through both muscle and nerve fatigue.

If the normal reactions are known before a test, in the case of any individual or group of individuals, similar measurements made at the end of, say, a 300-mile ride should show an appreciable difference. This difference, Dr. Moss thinks, may be a direct measure of comfort. The results obtained in a preliminary series of tests, using University students as subjects on rides of 300 miles over good roads and 150 miles over rough roads, were reported in the Automotive Research section of THE JOURNAL[1]. Further experiments have resulted in the selection of six tests which appear to be the most reliable.

[1] See S.A.E. JOURNAL, September, 1929, p. 298.

How Physical Fatigue Is Measured

Physical fatigue is measured by ascertaining the amount of chemical change in the body during a given period. In medical terms this is designated as change in basal metabolism and is measured by the oxygen content or volume of oxygen required before and after the ride. The apparatus used consists in an arrangement for having the subject breathe from a tank of oxygen, the amount consumed being measured by the fall of the line recorded on a revolving drum.

Physical fatigue is also measured by determining the decrease in the carbon-dioxide combining power of the blood. This test measures the ability of the blood to take up carbon dioxide. When the body becomes fatigued and the blood is taxed with an extra supply of carbon dioxide, the combining power of the blood decreases. Dr. Moss declares that this test gives as reliable results as may be expected in measuring a very small amount of muscular fatigue such as is acquired in automobile riding, and states that the test appears to be of value also in recording nerve fatigue. This test has given very consistent results, showing a definite decrease in the power of the blood to take up carbon dioxide with an increase of fatigue, and furthermore results so determined are little affected by emotions. However, it has the disadvantage of requiring 10 cc. of blood from each subject, which, although not objectionable from the standpoint of application in research work, would probably eliminate its use on motorcoach passengers. Its chief value, therefore, lies in providing a check for the validity of results obtained in the other tests.

Four Measures of Nerve Fatigue

Nerve fatigue is measured in four ways. First, by causing the subject to stand on a "wabble meter" which measures the body sway or steadiness. Readings are taken both before and after the ride, the theory being that steadiness decreases as fatigue increases. The first form of this device was arranged so that the subject had under the ball of his foot a bar which, if pushed down too far, allowed to swing up too far or moved from side to side by the person's losing his balance, makes an electrical contact and registers by ringing a bell or operating a counter. This test is a measure

(*Continued on p.* 101)

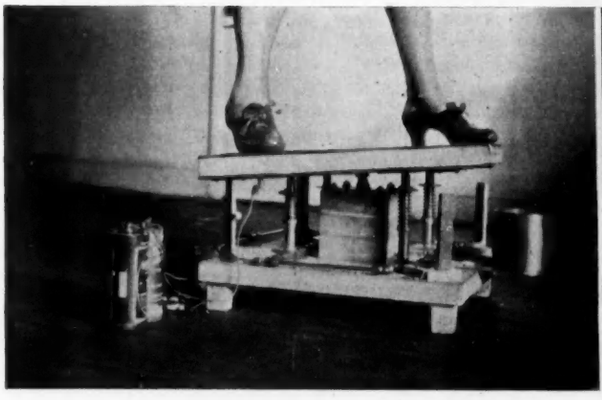


FIG. 1—ONE OF THE WABBLE-METERS DESIGNED AT THE BUREAU OF STANDARDS

The Subject Stands on the Platform, Which Is Mounted on a Universal-Joint so that Sway in Any Direction Makes an Electrical Contact That Operates a Recording Device

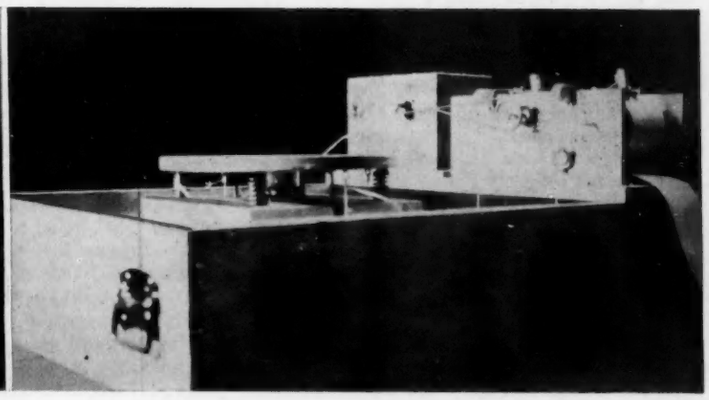


FIG. 2—WABBLE-METER HAVING PLATFORM MOUNTED ON A BALL-AND-SOCKET JOINT AND SUPPORTED BY COIL SPRINGS

Tilting of the Platform in a Fore and Aft Direction Operates Two Cords Attached to a Pencil Which Is Drawn Back and Forth on a Moving Tape Producing a Graphical Record of the Motion. A Second Set of Cords Operate with a Side-to-Side Tilting of the Platform

Standardization Progress

Aircraft-Engine Shaft-Ends

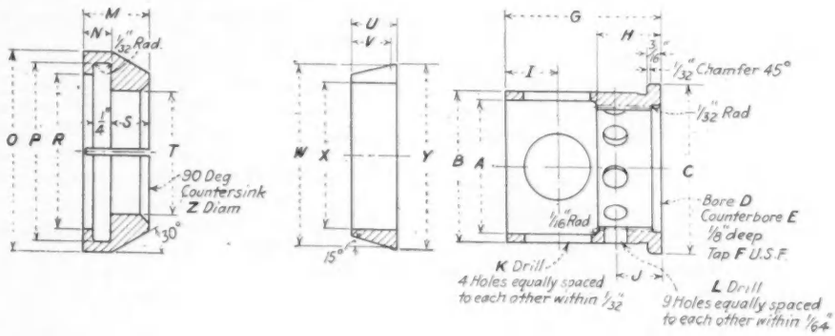
New No. 50 Shaft-End Dimensions and Cone and Nut Standardization Being Considered

AT THE time the present S.A.E. Standards for Propeller-Hubs and Shaft-Ends were developed, it was thought inadvisable and unnecessary to include any size larger than the No. 40. With the advent of larger engines of greater horsepower, the demand for a large splined shaft-end has increased and the question of desirable dimensions and tolerances for the proposed No. 50 shaft-end is again under consideration by the Aircraft-Engine Division.

Cones and Nuts

Some previous figures showing the proposed dimensions have been printed in THE JOURNAL. Since that time, however, further consideration has been given the proposal and a new table of dimensions submitted to the Aircraft-Engine Division. This matter will be brought before the Division meeting in St. Louis for discussion at the time of the International Aircraft Show in February. The proposed specifications are as illustrated on the opposite page.

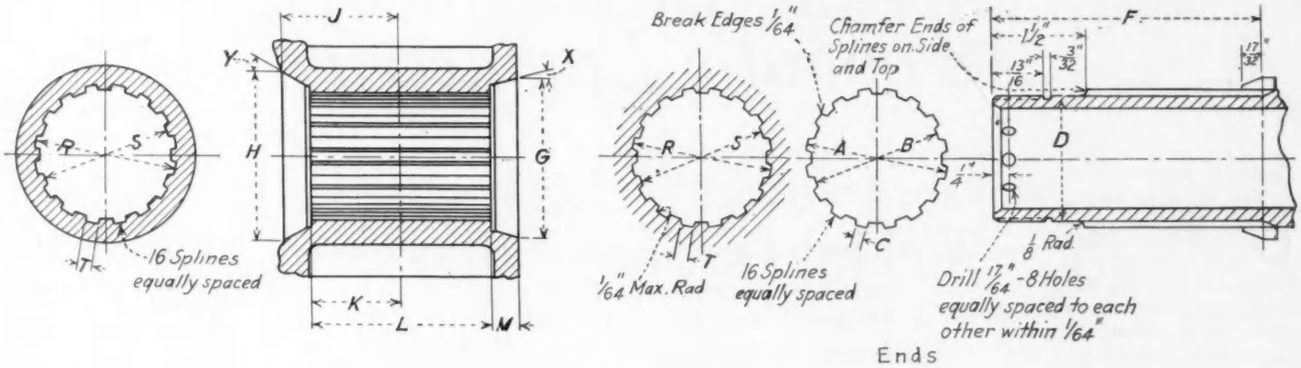
Ever since the adoption of Propeller-Hub and Shaft-End Standards, the Society has been requested to take up, through the Aircraft-Engine Division, the standardization of the cones and nuts used in mounting the hub on the shaft. The accompanying drawings and table show a proposed series of dimensions for these parts as they are to be submitted to the forthcoming Division meeting. Indications are that the servicing of propellers will be aided to a great extent if interchangeability of these parts on engines having shafts of comparable sizes can be accomplished. The comments and criticisms of the industry are invited so that such suggestions can be placed before the Division at its February meeting.



PROPOSED PROPELLER CONE AND NUT DIMENSIONS

S.A.E. No.	*A*	*B*	*C*	*D*	*E*	*F*	*G*	*H*	*I*	*J*	*K*	*L*
10	1-13/16	2-1/16	2- 5/16	1.601±0.002	1-45/64	1-11/16—12	2-1/8	7/8	23/32	5/8	57/64	5/16
20	2- 3/16	2-7/16	2-11/16	1.976±0.002	2- 5/64	2- 1/16—12	2-1/8	7/8	23/32	5/8	57/64	5/16
30	2- 3/8	2-5/8	2-15/16	2.226±0.002	2-21/64	2- 5/16—12	2-1/8	7/8	23/32	5/8	57/64	5/16
40	2-15/16	3-3/16	3- 7/16	2.726±0.002	3- 7/8	2-13/16—12	2-1/8	7/8	23/32	5/8	57/64	5/16
50	2-15/32	3-3/4	4- 3/16	3.351±0.002	3- 9/16	3- 7/16—12	2-7/8	1-9/32	27/32	1-1/16	1-1/32	3/8

S.A.E. No.	*M*	*N*	*O*	*P*	*T*	*U*	*V*	*W*		*X*	*Y*	*Z*
10	29/32	13/32	2-3/4	2- 7/16	1.6875±0.0005	5/8	17/32	2.500		2.001 ±0.0005	2-17/32	1-15/16
20	29/32	13/32	3-1/8	2-13/16	2.0625±0.0005	31/32	17/32	2.875		2.3760±0.0005	2-15/16	2- 5/16
30	29/32	25/64	3.250 ±0.005	3	2.3120±0.0005	25/32	17/32	3.187	+0.005 −0.000	2.626 ±0.0005	3- 1/4	2- 9/16
40	29/32	13/32	3-7/8	3- 9/16	2.8125±0.0005	23/32	17/32	3.625	+0.005 −0.000	2.126 ±0.0005	3-11/16	3- 1/16
50	1	7/16	4-9/16	4- 1/4	3.4375±0.0005	25/32	5/8	4.625	+0.005 −0.000	3.8130±0.0005	4-11/16	3- 3-16



PROPOSED NO.-50 HUB DIMENSIONS

Hub No.	Front End	Rear End							For Rear Cone	For Front Cone
	H	*G*	*J*	*K*	*L*	*M*	*R*	*S*	*T*	*X*	*Y*
50	$4.562^{+0.005}_{-0.000}$	$4.625^{+0.005}_{-0.000}$	2-13/16	2-5/32	4-5/16	3/4	$3.820^{+0.005}_{-0.002}$	$3.568^{+0.005}_{-0.002}$	0.377 ± 0.001	15°	30°

PROPOSED NO.-50 SHAFT-END DIMENSIONS

Shaft No.	Shaft-End								
	A	*B*	*C*	Thread	*F*	*R*	*S*	*T*
50	$3.812^{+0.000}_{-0.002}$	$3.562^{+0.000}_{-0.005}$	3.750 ± 0.0008	3-7/16—12 U.S.F.	P.D. $3.437^{+0.000}_{-0.002}$	6-31/32	$3.820^{+0.005}_{-0.002}$	$3.568^{+0.005}_{-0.002}$	0.377 ± 0.001

Automotive Research

(*Concluded from p.* 99)

of coordination between nerve and muscle but is principally a measure of nerve fatigue. The principle involved seems most promising, but the device is being improved upon. Several instruments of this type are being developed and tried out at the Bureau of Standards in cooperation with Dr. Moss's laboratory, and it is expected that a "wabble-meter" with the necessary stability will be available this month. Figs. 1 and 2 show two of the later devices now being tried out.

Dr. Dickinson of the Bureau of Standards has also devised a tubular form of steadiness indicator which tests the steadiness of the hands and is giving reasonably good results. However, in general, from tests tried on the hands, it appears that they are not as noticeably affected as are other parts of the body.

A third method of measuring nerve fatigue which has proved satisfactory is a system of number checking. Printed sheets containing 1000 numbers, 100 of each of the 10 digits mixed together at random, are placed before the subject and it is ascertained how fast and how many of a particular figure, 5 for example, will be checked in 1 min. With mental fatigue a decrease in the ability of the subject to maintain his pre-ride average is noted.

The fourth test, selected for its consistency and ease of application, is mental multiplication, that is, the rate at which it can be done, before and after the ride. It is true, Dr. Moss pointed out, that this test shows a tendency of the subject to improve with practice but only up to a certain point, then riding produces a falling off in performance. However, a new subject going through the tests for the first time might improve sufficiently between trials to offset the falling off due to fatigue.

More Extensive Tests Warranted

In the main, it may be said that all of these tests are in the process of being improved but enough information has already been obtained from the research experiments to warrant extensive work along this line. The need for developing methods that will eliminate the tedious working up of records and substitute a counting or integrating attachment is recognized.

Furthermore, when the work has progressed to the stage at which an attempt can be made to correlate the qualitative results obtained in the laboratory with the impressions of relative comfort orally expressed by the passengers, the effect of factors other than vibration must be considered, such as color scheme, interior finish, action of the windshield wiper, and other apparently minor details.

Admittedly, human physiology and psychology play a leading rôle in determining satisfaction or dissatisfaction with a ride. In these experiments, therefore, the human body is being used as a seismograph, with the hope that the readings, though rough in form, may indicate the way in which instruments should be used for measuring riding-comfort. To the automobile manufacturer who is interested in increasing the comfort and luxury of his product, these tests are expected to supply data of considerable value.

Dr. Moss will present a second progress report on these experiments at the Riding-Qualities Session of the forthcoming Annual Meeting of the Society.

Aeronautic Engineering

Welded Fuselages Criticized

Army Airplane Maintenance Officer Would Prefer to Repair Structures Without Welding

COMMENTS upon the subject of welded steel-tube fuselage structures have been received from Hugh C. Downey, first lieutenant, Air Corps, who is chief engineering officer at the French Field Air Depot, Canal Zone, Panama. Lieutenant Downey's remarks follow:

Difficulties with Welded Structures

Various people have been trying to impress the industry with the manifold advantages of the welded steel-tube structure for the fuselages of airplanes, and J. H. Kindelberger tries to clinch the argument with his interesting article in the S. A. E. JOURNAL for November, p. 474. Nevertheless, I beg to differ slightly.

While it is true that the welded steel-tube structure is rigid, simple and probably amply strong, the experience of several years in the Canal Zone and other places has pointed out several shortcomings.

The first and probably most exasperating difficulty with welded steel-tube structures is that they require rigid jigging to prevent distortion during manufacture or repair, and that they do distort, even with that. Suitable allowances for fusing and shrinking of welds should correct this, but they do not.

The second difficulty is that, once welded to another, no part can be replaced without much filing, cutting, fitting and rewelding.

The third difficulty is that heat-treating the completed structure is impracticable, because of its bulk.

Some aircraft engineers make the great mistake, which Mr. Kindelberger's company does not make, of using welded steel-tube diagonals in fuselages. Such diagonals should not be used, because they allow no adjustment; and an airplane fuselage having no provision for adjustment of alignment is very poorly designed.

Many of these modern welded steel-tube structures, as Mr. Kindelberger points out, have complex fittings, welded to form one piece, with several lugs jutting out at odd angles. This is also undesirable, since the failure of one lug may necessitate the replacement of a large, expensive part. A pin-jointed structure, or possibly a riveted structure, so built that minor parts can be replaced without cutting or welding, is much better than a welded structure.

Welds, especially electric welds, often show cracks due to either fatigue or initial and unequal stresses from welding operations. The only cure for these is to file away the old weld and reweld, possibly with a reenforcement, with consequent annealing of the metal adjacent to the weld. Parts of a riveted or pin-jointed structure can be replaced easily, and the various parts of a pin-jointed structure can be heat-treated easily, if desirable.

Difficulties in Field Repairs

I recall a certain pursuit airplane in which the tail-skid post rotated in a steel bushing which was welded to the fuselage structure. This particular part showed an incurable tendency to crack at the welds and pull away from its supporting members, some half-dozen of which had to be cut away and replaced when replacing the bushing. In spite of heavy jigging, it was necessary to ream the new bushing after welding it in place. If the part had been bolted in, this would have been unnecessary.

Landing-gear lugs on the same plane required microscopic inspection very often, because of the great tendency of the metal to crack away from welds under fatigue. This need for exhaustive inspection during overhauling, to be certain that everything is as it should be, is perhaps the greatest objection to welding.

I recall also a certain observation plane that had a spring-suspended tail-skid. The bolt that held the spring in tension began to break, so the maintenance people did the obvious thing and strengthened the bolt, whereupon the welded tubes supporting the swivel of the tail-skid began to crack off. The next step was to gusset these tubes so that they would not crack away. Finally, the longitudinals began to show signs of failure, so the disintegration seemed to be without end.

The trouble in this structure seemed to be due to lack of sufficient suspension at the spring, which caused sharp impact. Also, there undoubtedly was unequal initial stress on the several tubes that supported the swivel post. In fact, unequal initial stresses are believed to exist in all welded assemblies, because of unequal expansion and contraction. Much trouble is traceable also to insufficient area under the weld and to lack of adequate gusseting.

Fire Hazard in Welding

Another source of worry is that most airplanes undergoing minor repairs which require welding are inflammable and usually have gasoline in the tanks. The greatest precautions are necessary to prevent fire. Maintenance officers no doubt shorten their lives considerably by worrying about the problems of welding.

I suspect that the chief reason for the great vogue of the welded fuselage is that welding is a rapid manufacturing process in a well-organized plant; but it is anything but rapid in the field. Welders can be trained without great difficulty, but the welding equipment is bulky and hard to transport, so that usually, when an airplane develops cracked welds or cracks adjacent to welds, no choice is left but to fly it to a depot in a gingerly manner.

After having the responsibility for the repairing of a number of such planes, I would heartily welcome some other type. It is realized that bending movements are greater in a pin-jointed structure than in a welded one, but this would be prevented in a properly designed riveted affair. The number of parts would be greater, but screw-machine parts are easily manufactured. I incline toward foreign practice, in which only minor parts that are not heavily stressed are welded.

Demands of Machine-Tools

W. P. Michell Tells Requirements and Experiences of a Large Maker of Commercial Cars

DISCUSSING a variety of subjects relating to machine-tools, during the Production Forum at the recent Production Meeting in Cleveland, W. P. Michell, of the International Motor Co., Allentown, Pa., outlined the principles under which machine-tools are selected by a large manufacturer of motor-trucks.

One of Mr. Michell's chief demands is that the machine shall be as nearly automatic as possible, requiring the operator only to insert the part and start operation by a lever. He measures present machine-tools by this standard and gives several advantages of the economy resulting from the substitution of machines requiring less attention on the part of the operator. Chip removing and handling and burring of the finished parts are improvements demanded.

These points were supported by discussers, who also entered into the merits and demerits of hydraulic operation of feeds and controls and the uses and limitations of antifriction bearings. A concrete suggestion was made for an improved method of handling chips. Mr. Michell's paper follows:

Requirements of a Truck Manufacturer

Standard machine-tools will continue to predominate over special machinery, I believe, and true economy in purchasing will be attained by allowing machine-tool builders to furnish standard machines fitted to the job in hand and subject to specifications. Machines for special purposes tend to become standard. In purchasing a machine for a particular job, we generally ask bids from the machine-tool builders best qualified to figure on the machine and tooling, and our decision is based on the following factors:

Dimensions and weight; the more rugged machine is preferred, other things being equal.

Durability; the machine having antifriction bearings will be strongly favored over one with plain bearings. On production lathes, either turret or double-cross-slide, we believe hardened and ground steel ways to be absolutely essential. We had a center lathe with gang tool-blocks front and back, furnished by one of the leading makers, which went out of commission in six months because of wearing and scoring of the slides on the chilled-iron ways.

Lubrication may be of the type best adapted to the machine; splash, pressure-gun, central-pump, or gravity-manifold with tubes to the bearings are considered desirable types. Individual oil-holes, requiring the operator to oil his machine each morning, are not desirable and are most frequent causes of bearing trouble.

Automaticity. I believe that the only functions of the operator of the modern machine-tool should be to load, start and unload the machine.

Production estimates are given consideration only as they bring out features not otherwise considered.

Heat-treated alloy-steel gears running in oil are fortunately almost universal on modern production machinery, so usually are not a factor in the decision.

Automaticity of Present Tools

Let us see how well the present machine-tools come up to this standard of automaticity; that is, that the operator's only function should be to load, start and unload the machine.

The hand-operated sensitive drill will give way to hydraulic or friction-feed machines with rapid traverse to and from the work.

The ordinary turret-lathe will be replaced by the semi-automatic; this change, it seems to me, has been too long in coming. The final costs will usually indicate the semi-automatic machine for new jobs or additional capacity, but replacing existing equipment must be considered on its own merits for each case.

The drill-press will be equipped to traverse rapidly to the work and feed through automatically, as well as to trip and raise the spindle as at present.

The new cylindrical grinders are now in line with this rule. The operator loads the work and throws the starting lever; the head-stock starts and stops; the flow of grinding compound starts and stops; and the wheel traverses rapidly to the work, grinds it to size, and returns, all automatically. These machines compete successfully with centerless grinders.

Internal grinders are now made automatic except for loading and starting.

Milling-machines of certain types are equipped with these automatic features; but nothing of the sort is available on the hand-miller type, where the saving would be greatest.

Disc-grinders which require traversing of the work by a hand lever should be replaced to a considerable extent by the mechanical or hydraulic power-traverse disc-grinder.

The tool and cutter-grinder in which the operator indexes the cutter by hand will probably become obsolete in larger-production shops.

On thread-millers, tapping-machines, bar-pointing machines, centering machines, and bolt-threaders, the operator must usually traverse the work or the cutter. Machines which eliminate this hand traversing are above competition, other things being equal.

Savings from Automatic Operation

I have some figures which illustrate the saving resulting from the elimination of hand machine-operation.

The labor cost to turn and bore a steel casting on a heavy-duty turret-lathe was $1.48. One man now runs three automatic turret-lathes at a labor cost of 35 cents per part. Each of these machines is tooled for several different jobs.

On a semi-automatic plunge-cut grinder, the time for grinding a spindle is 1 min.; against 1.2 min. on a centerless grinder and 5 min. on a traversing grinder. This saving is partly due to a wide wheel on the large machine and partly to automatic features which relieve the operator, so that he measures the last piece and places the driving-dog on a new piece while the machine is operating.

Two center lathes of standard type were bought, some years ago, in which the slides are traversed to and from the work by a hand-wheel. Because of the time this required, one operator was needed on each machine, instead of one operator for the two machines.

On a friction-feed sensitive drill with automatic spindle-traverse to the work and return, production is 1000 per hr. against 600 by hand-operated sensitive drill. Breakage is less on the friction-feed drill.

We formerly used a very heavy-type adjustable multiple-spindle drill for drilling ten 1 5/32-in. holes in a forging which is heat-treated to a high Brinell reading. The life of a drill between grindings averaged about eight holes, at a cutting speed of 20 ft. per min. and

a feed of 0.0024 in. per revolution. The cutting speed on a large hydraulic-feed single-spindle drill equipped with solid non-adjustable multiple heads is 27 ft. per min., with a feed of 0.0078 in. and an average drill life of about 15 holes per grind. In this case, the more rigid machine doubled the drill life; also, the labor cost was greatly reduced, because of the automatic hydraulic operation and the higher speeds and feeds possible with the more rigid machine and hydraulic feed.

Chip Separation and Burr Removal

There is no excuse for dropping the work of an automatic screw-machine into the pan with the chips. The cost of separating the work is considerable, even when a special machine is used for the purpose.

Not enough attention is paid to chip handling. Chips should fall into a hopper which can be removed bodily from the machine and dumped. The practice of raking chips out of machine pans is wasteful.

Why should not the automatic screw-machine be made with an attachment to remove the cut-off burr from each piece? Automatic machines are available to remove this burr from screw heads; on other turning work, the cost of burr removal may be more than the turning cost.

THE DISCUSSION

E. P. BLANCHARD[1]:—Builders of machine-tools realize that no automatic machine can exist in the present system of production without automatic lubrication. This has gradually become an absolute essential of automatic machinery since its introduction in 1907.

W. J. F. FORWARD[2]:—I have had the opportunity to learn what are some of the things demanded of special high-production machine-tools, in the course of manufacturing a large quantity of such tools. Hydraulic feeds are being demanded in almost every case and hydraulic mechanism for as many functions as possible, including hydraulically operated clamping devices and index pins. Almost every hydraulically operated function is satisfactory to the purchaser. Hydraulic feeds are desirable because of the flexibility of their regulation, ease of control and long life.

We are being asked for tapered-roller bearings on spindles, and some form of antifriction bearing on other parts of the machine. The tapered-roller bearing maintains correct spindle alignment indefinitely. The side thrust on the spindle bearings tends to push them out of alignment with the table. On this account, plain bearings need to be replaced or rescraped, but roller-bearings need only to be adjusted.

Special machines are wanted as compact and solid as possible, to reduce the overhang and to make possible heavy feeds. We are also required to make special machines that can be changed to accommodate similar work on parts of various sizes. We do that by using straight box sections. The result may not always be beautiful, but the straight box section can be shortened and lengthened very easily in making a second machine of the same type for a larger or a smaller job.

Other things that are being asked for in automatic machines are either one-shot or pressure-pump lubrication or automatic lubrication by a pump inside the machine; automatic cycles of operation; automatic clamping, so that the operator does not need to check several locating points, possibly with gages; and the performance of as many operations as are economically possible on one machine. Convenience of set-up becomes more important as the number of operations on one machine increases. Separate machines for each operation are better than three or four operations on one machine if the setting-up is not very convenient.

A MEMBER:—Additional advantages of the hydraulic feed are smoothness of work and longer life for cutting-tools.

Criticism of Hydraulic Feeds

ROBERT R. KEITH[3]:—We have had unfortunate experience with hydraulic feed in tools of a number of different types. The reason is the variation in feed resulting from the change in viscosity of the oil at different times of the day. This makes it difficult to set a fixed rate of feed and therefore a fixed piece-work rate. Nevertheless, hydraulic feed is supreme in a very definite field. Included in that field are the control of the platen of internal and larger grinders of the diamond type and the reversing feed used in crankpin grinding.

Tapered-roller and other antifriction bearings also have a very definite field in which nothing else is quite so good; but, for certain types of work, particularly in cast iron, the large well-lubricated plain bearing is still hard to excel. Hardened and ground bearings operate better than any others that have been found so far in certain kinds of high-speed work also. We must guard against being hypnotized in our designing by any idea of the universality of any one kind of bearing, feed or any other device.

MR. FORWARD:—The variation in feed of a hydraulic system resulting from the change in viscosity of the oil is considered serious by some manufacturers, particularly those who require accurate timing in their production lines. Two or three large users of tools in the neighborhood of Detroit will not consider hydraulic machines for that reason. The difference in production rate between early morning and later in the day may be as much as 25 per cent, and this becomes serious in a plant running at a high production rate with no storage space for parts.

Handling the Work and Chips

LOUIS A. SCHAAFF[4]:—The company with which I am connected produces a chromium-plate of exceptional density and has concentrated its efforts on producing plated surfaces for industrial purposes, particularly for antifriction surfaces. In this way we have so largely reduced the friction and the need for lubrication of a textile spinner-ring, the flyers of which operate at 4800 r.p.m., as to prolong the service of the flyer from 6 hr. to nearly 6 weeks. The reduction in the amount of lubrication bids fair to eliminate the staining of the material by the lubricant carried on the flyer, so that the material may no longer need to be washed or redyed. We are also experimenting on slow-moving machinery with the object of eliminating lubrication. From what we have found, I believe we can produce a surface so smooth that, if it were used for a roller-bearing race, the coefficient of friction would be so low that the rollers would not roll.

MR. FORWARD:—I have been studying the question of chips for many months and believe I have a practical solution. That is, to adopt a standard size of chip-pan to be placed under all the machines in the plant. These pans should be elevated from the floor to allow the use of a lift-truck. All the chips under a long machine can be caught by a single pan by providing a belt conveyor underneath the machine, emptying into the chip-pan.

[1] M.S.A.E.—Assistant sales manager, The Bullard Co., Bridgeport, Conn.

[2] Sales and development engineer, Consolidated Machine Tool Corp. of America, Rochester, N. Y.

[3] M.S.A.E.—Manager, Tractor Works, J. I. Case Threshing Machine Co., Racine, Wis.

[4] Director of sales, Chromeplate, Inc., New York City.

Transportation Engineering

Tractor-Semi-Trailer Operation

A. W. Kenerson Outlines Means Whereby Interchangeability Was Accomplished

A CAREFUL study of the application of the semi-trailer to the petroleum - transportation problem from the operator's viewpoint reveals many possibilities not apparent on the surface, and it is with this phase in mind that A. W. Kenerson, of the Standard Oil Co. of Ohio, has amplified the discussion he offered at the Toronto Transportation Meeting to show the benefit derived by his company's analysis in connection with the various items of the subject.

It may seem far-fetched, says Mr. Kenerson, to have a tractor in Cleveland, regardless of make, interchangeable with trailers throughout the entire field of operation, but this opens up great possibilities from the viewpoint of flexible and continuous operation, which is comparatively easy to accomplish if the specifications are set up with that idea in mind. To that end, the development of the light tractor and semi-trailer unit as used by his company began with specifications for the trailer.

The application of tractor-trailer automotive equipment to the transportation problem had its inception in the desire on the part of an operator to move more pay-load than could be moved on a truck of the maximum capacity then available, according to Mr. Kenerson. This resulted in the development of the tractor-trailer combination which made it possible to move several times the rated capacity of the tractor, and, although several compromises were involved in this practice, it proved economically sound.

Semi-Trailer Specifications Developed

Continuing, Mr. Kenerson states that the specifications developed were for a semi-drop-frame trailer designed to be applicable to a dozen or more different tractors, a list of which was submitted to the engineering department of the trailer manufacturer and included tractors of extreme heights and widths to make certain that the trailer would be universal in its application to any tractor. The trailer has a 110-in. wheelbase and is fitted with manually operated jacks, which his company considers a very necessary feature.

The specification for wheels constitutes one of the compromises incidental to the use of the light truck as a tractor and should be given careful consideration. The tractor originally designed to move a gross weight of approximately 4000 lb. including chassis, body and pay-load is now being used to move a gross weight of approximately 12,000 lb. But, while it is true that the tractor carries only one-third to one-half of the *pay-load*, due consideration must be given the starting and stopping of this increased gross-weight because gear-tooth pressures are stepped up and the wheel stress in general and the shear on the wheel studs are increased. This applies not only to the tractor driving-wheels but to the trailer wheels also, on account of the use of power brakes.

The company therefore deemed it necessary to obtain for heavier duty a wheel having a larger bolt circle and larger-diameter studs than are furnished as standard equipment by the manufacturer, at the same time being careful to avoid the selection of a wheel that might be too heavy for the front-axle steering mechanism. Further, it desired to obtain a trailer wheel that would be interchangeable with one or more makes available on the various tractors, thus obtaining a second source of supply. In reaching this conclusion, a work sheet was used on which were listed the manufacturers' specifications for the various tractors. Both standard and optional wheel-equipment were shown, to make evident the more universally used equipment and thus aid in deciding upon the desired specifications.

It was believed that the tractor-brake maintenance could be reduced to the minimum if the trailer brakes, which are operated by a vacuum booster, were of sufficient dimensions to provide for more than half the braking. This is also beneficial from the viewpoint of tractor-frame stress-reduction as compared with having the tractor do most, or all, of the braking.

The tank specifications were constructed around the limiting factors set up by the capacity desired and the trailer dimensions. These tanks have four compartments, fitted with emergency-valve equipment of various types.

The tractors for petroleum-industry operation should be fairly fast and have the necessary acceleration to maintain their place in traffic. An endeavor was made to obtain tractive effort sufficient to negotiate a 3-per cent grade with the maximum gross weight in direct gear and with rear-axle gear-ratios that would make possible a road speed of 30 m.p.h. at approximately 2000 r.p.m. of the engine.

The cab is an important part in that it provides the driver with comfortable protection against the weather; expense incurred in this direction by the purchaser is very definitely reimbursed by the better care the driver gives the equipment. Upholstered de-luxe-type cabs are not warranted, but it is possible to obtain a fully enclosed cab having de-luxe lines, that is comfortable and roomy, and that affords clear vision. This type has stood up well.

Adaptation of Tractors to Trailers

The company's next procedure was to adapt the various tractors to the tank-carrying trailers. This was comparatively simple because of the provisions made in the original design of the trailer. Each fifth-wheel hook-up was worked out from the actual assembly with the tank loaded, so as to obtain correct levels. Variation in spring deflection had to be taken into consideration. A record of these assemblies has been made in chart form, for future use, which shows the following fundamental dimensions: chassis wheelbase, distance back of cab to center of rear axle and to center of fifth wheel, height and width of frame, location of vacuum-brake connections and height of wood shim under the fifth wheel.

The simplicity of the universal application of the various tractors to the trailers is indicated by the fact that the characteristic differences of these assemblies per make or model of chassis are confined entirely to fifth-wheel location and height of the shim under the fifth wheel.

Further simplification of the problem is indicated in that the 1000-gal. tank shown in the accompanying illustration is mounted on the same trailer chassis as is the 750-gal. tank. The only detail changed is the tire specification, changed from 30 x 5-in. to 32 x 6-in. dual pneumatic tires. The wheel selected provides correct spacing for the 6-in. tire. There is a slight compensation of weight in this set-up, because the 1000-gal. unit is used as a small-town service-station truck; the

drum carriers being omitted, which reduces the weight 800 to 900 lb.

The company also operates several semi-trailer drays of 15-bbl. capacity, the fifth wheels of which are designed to hook up with these same tractors. In fact, it has two or three installations in which one tractor is operating as a shuttle unit between a tank trailer and a dray trailer.

Allocation of Equipment

The problem of properly allocating this equipment in the field was influenced by the somewhat varied tractive ability of the tractors and differences in the operating requirements. As definite information was required, a numerical designation of these operations was set up as follows: No. 1 is an improved-road operation, 5 to 10 per cent of which might be on grades; No. 2 is on improved roads, 50 per cent of the operation involving grades; No. 3 is an operation 75 per cent of which requires the negotiation of grades; and the No. 1 and No. 2 operations may be on unimproved roads. The company's territorial men were requested to specify on their requisitions for new equipment the type of operation in which it was to be used, and thus the equipment was placed in the territory to the best advantage.

Operation Flexibility Secured

Flexibility of operation is made possible by this universal application of tractors of the various makes to tank and dray trailers throughout the State. One of the tractors was laid up for 10 days as the result of an accident which made necessary the replacement of the front axle, the frame, and the engine bell-housing. It was operating as a shuttle unit between a 3-ton dray and a 1000-gal.-tank trailer; but the trailer, the pay-load end of this equipment, went back into service the next morning because another tractor could be dispatched to that location.

The field mechanics are provided with a Ford tractor equipped with a fifth wheel and the customary field-maintenance tool-equipment. To take a piece of equipment out of service during the peak season for any extensive maintenance operations is very expensive. Too often a driver will report that the valves require attention, the rear axle is noisy, and the brakes need adjusting, and in the same breath will request that the brakes be "fudged up" and he be allowed to continue in operation because he is already behind his schedule. The danger of continuing the tractor in service is that the operator thus courts more extensive repairs by allowing valves that do not seat properly or have proper clearance to continue in operation, and that a noisy rear axle very often results from improper gear-adjustment that will cause complete failure if continued in operation. But when a field mechanic equipped as described arrives, the local driver detaches his tractor and substitutes the Ford tractor, which functions as a conveyance for the field mechanic and as a temporary tractor with which the driver can continue on his schedule, after having been interrupted only for the time required to make the tractor transfer.

This procedure makes it possible to maintain very complete field-maintenance operations regardless of the time involved. Further, it is possible for the field mechanic to function as a preventive-maintenance man, charged with the responsibility of keeping the equipment out of the shop and on the road. This arrangement, together with the complete replacement of a tractor, provides that continuous operation which has so much to do with reducing the hauling cost per gallon.

This same procedure becomes a very important requirement in the bulk-haul program, as the uninterrupted operation of these trains is a vital economic factor, because a greater investment that increases interest and depreciation charges is involved. One of the great economies realized by this type of operation is the reduction of bulk station-transfers by direct service-station deliveries. This necessitates a very dependable service which can be maintained only by very close adherence to a schedule; in other words, time out of service must be reduced to the absolute minimum. To accomplish this, the 2500 or 3000-gal.-capacity semi-trailer design must be universal in its application to the various heavy-duty tractors available.

Extra Heavy-Duty Tractor Justified

The operation of six or more of these trains easily justifies the investment in an extra heavy-duty tractor to migrate throughout the territory manned by a specialist in maintenance of this type of tractor. To facilitate the movement of this unit at fairly high speed when traveling light between locations, two-stage springs and an over-geared transmission would be practical considerations. The operation of such a unit provides for regular roundhouse inspections and proper field maintenance with no interruption of train schedules whatever; also, complete insurance against tie-up due to emergency.

The company's experience with this type of equipment justifies the mention of several other features. In regard to handling, it is easy to maneuver in and out of the most inaccessible service stations on account of its small turning radius; in fact, it has been found possible to turn one of these trains around on a 14-ft. road, because of the short wheelbase of the units used. This type of construction also has a very low center of gravity, which makes the equipment safe to handle at high speed on curves, since it has no tendency to sway sidewise.

Mechanical Flexibility and Operating Safety

Mechanical flexibility is provided in all directions by the fifth wheel at a very critical point in the construction. One of the serious problems the operator of the four-wheel tank-truck has to contend with is frame breakage. This has been found in 90 per cent of the cases to be the result of mounting a tank, which is a rigid member, on a frame designed to flex under road-variation stresses. This practice localizes the flexing of the frame at a point between the cab and tank. This flexing stress is eliminated entirely by the tractor-trailer flexibility.

Having the tank mounted on a semi-trailer removes it farther from the exhaust pipes and other heated surfaces, thereby reducing the fire hazard. The outfit is equipped with six wheel-brakes; the tractor manufacturer provides four-wheel brakes, and the trailer is equipped with B. K. booster brakes

TRACTOR-SEMI-TRAILER EQUIPMENT USED BY THE STANDARD OIL CO. OF OHIO
In the Several Different Types of Service Throughout This Company's Territory, All the Tractors and Trailers Are Interchangeable

that operate simultaneously with the tractor brakes. This brake equipment makes possible the operation of the unit at high speeds with the increased safety that is so necessary in congested areas.

In regard to maintenance, the trailer can be uncoupled from the tractor and run into the shop as a separate unit, thus increasing the productivity of the floor space because much less area is occupied. The rear axle and brake mechanism are all exposed, which greatly facilitates brake adjustment and lubrication.

Assembly of Units and Initial Cost

The possibility of putting this equipment through the shops as separate units during the process of original painting and assembly has facilitated and simplified the shop program. The trailer, with a tank mounted on it, can be in process of being painted and lettered while the tractor is in the blacksmith shop, where the cab and the fifth wheel are mounted. This, together with the fact that these units are numbered separately, makes it possible to hook up a finished tractor with a finished trailer and send it into the territory with the least possible delay.

As to initial cost, the problem as stated by Mr. Kenerson is to move a certain gallonage with a limited budget available for new motor-vehicle equipment. When the fact is considered that it is possible to increase the capacity 100 gal. at an initial saving of $800 to $1,000 per unit as compared with the average 2-ton-capacity truck, the advantage is obvious.

All through the specifications great care has been taken to avoid anything whatever that might complicate the situation for the manufacturer. An endeavor has been made to have this unit as practical as possible from the production viewpoint as well as from the operating viewpoint. The hope of the company has been that other operators would find it desirable to use this unit, thereby developing a production volume that would have a beneficial effect on the cost, which obviously has its appeal to the purchaser.

Welding Procedure Described

Orlie Trentham Summarizes Practices Used in the First-Class Repair-Shops

A RÉSUMÉ of the welding processes applied to automobiles and motor-trucks, as practised in some of the first-class shops in this Country, was included by Orlie Trentham, of the Linde Air Products Co., in the paper he presented at a meeting of the S.A.E. Club of Denver, Colo. In his opinion, serious consideration should be given to correct procedure so that satisfactory applications of oxy-acetylene welding will be made and effective methods used.

Should the part in question be preheated and welded with cast-iron, welded with steel, or bronze-welded? Should it be welded in the car or removed for welding? he asked. We know that fenders, fender brackets, bumper and body parts can be welded in the car in most cases, he said; but cylinder-blocks, crankcases, oil-pans, starter housings and timing-gear covers must be removed in most cases, and the welding procedure depends entirely upon the nature of of the fracture or damage.

Mr. Trentham said further that, as proved by experience, cracked water-jackets on both cylinder-blocks and cylinder-heads should be bronze-welded, the cylinder-heads being removed for this work. Sometimes cylinder-blocks can be welded in the car, but generally should be removed. Cylinder-blocks with scored cylinder-walls should in all cases be removed, preheated and welded, cast-iron welding-rod being used in such case. The same applies to blocks or heads having cracks around or through the valve-seats.

Years ago, considerable trouble was experienced in preserving the alignment of starter housings welded with cast-iron welding-rod, Mr. Trentham continued, but they are now bronze-welded much more cheaply and more quickly. If they are carefully handled, proper alignment is maintained. Cast-iron flywheel-housings, timing-gear covers and crankcases having fractures extending to the outer edges, formerly welded with cast-iron, are at present being bronze-welded. Some are being done in the car, but the majority are removed. Crankcases and cylinder-blocks having broken engine-supports are usually removed and welded with cast-iron or aluminum, as the case may be. Aluminum crankcases, transmission cases and oil-pans are in all cases removed from the car, as they require a certain amount of preheating to remove strains and distortion. Fractures developed in rear-axle housings are generally brazed, the work in most cases being done on the car. This method eliminates heating to a high temperature. All brake and brake-drum supports are being bronze-welded. Frames in all cases should be welded with a high-test rod and, in exceptional cases, a reinforcing plate on the bottom of the frame should be used.

Welding Small Parts

Regarding the smaller, removable parts of automobiles, Mr. Trentham finds the oxy-acetylene process applicable to welding intake, exhaust and water manifolds. Most of these are of cast iron and are welded with cast-iron welding-rod. Carbureters and accessories come in for their share of brazing. Broken gearshift levers are welded and built up on the transmission end when badly worn and useless. Door locks and hinges can be brazed, these parts being made mostly of malleable iron. Tire rims with broken or lost lugs are welded or built up, as the case may be.

Disc wheels sometimes develop a circumferential crack, said Mr. Trentham, and these are always welded with high-test steel rods. Steel and aluminum bodies are made as good as new by means of the oxy-acetylene welding torch, as are body braces which break from time to time. Water-pump housings are reclaimed by brazing. Water-pump shafts, when badly worn, can be built up and turned down to correct size. It should not be forgotten that broken crankshafts can also be successfully welded if a suitable jig is used to assure correct alignment.

In addition to welding the parts mentioned and numerous others, Mr. Trentham said that the oxy-acetylene torch is being used for straightening bent frames, axles, brackets and the like. About the only part of an automobile that should not be welded is the steering-knuckle. This is a very vital part and there is too much danger that an inexperienced welder will make a poor job by applying too much heat, changing the structure of the steel.

News of Section Meetings

(*Continued from p. 13*)

fore the fuel reaches the inlet manifold, the following advantages are obtained:

(1) Cross-sectional areas of the manifolds can be increased to sizes which will reduce the velocities of the air-fuel mixture to such values that the pumping losses of the manifold will be minimized.

(2) A much more uniform mixture can be obtained; thus assuring excellent distribution to the several cylinders and consequent smoothness of engine operation at all speeds and loads.

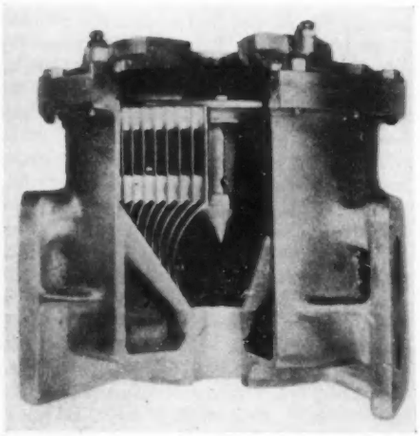
Some of the well-known fundamental principles governing the vaporization of a liquid were given by Professor Autenreith, who then described how the carbureter nozzle breaks up the liquid into a spray consisting of vapor, fog and droplets. He outlined the various methods used in vaporizing liquid fuels and then centered attention on a device with which the authors have experimented.

Vaporizing Device Described

Continuing, Professor Autenreith said that the device was designed to vaporize liquid fuels between the time they leave the carbureter and enter the inlet manifold. As shown in the illustration, it consists of a heating chamber of large volume and ample surface. The chamber is heated at the bottom by exhaust gases from the engine and the heat supply can be effectively controlled. The carbureter flange is attached to a mating flange on the bottom of the chamber and a passage is provided from the carbureter to the interior of the chamber. Resting on the bottom of this chamber is a nest of thin curved plates, arranged radially on a central vertical core and surmounted by a circular baffle-plate. The top of the chamber is connected to the inlet-manifold system of the engine.

The mixture as delivered from the carbureter enters the vaporizing chamber, where every particle is directed by the plates to move in a curved path. The globules or droplets of fuel cannot follow this curved path because of their inertia and because the velocity of the airstream is no longer sufficient to carry them with it, since the effective cross-sectional area of the vaporizing chamber is many times that of the carbureter and transfer passage. These liquid globules are, therefore, deposited upon the plates and sides of the chamber in the form of a thin film.

As the bottom of the chamber is heated by the exhaust gases and the lower edges of the plates rest upon this heated surface, the plates are heated also and are warmer at the bottom than at the top. Therefore all the constituents of the fuel in the film



DEVICE FOR VAPORIZING FUEL BEFORE IT ENTERS THE INLET-MANIFOLD

on the lower portion of the plates are quickly vaporized. The plates are warm enough near the top and have sufficient surface area to vaporize the lighter constituents of the fuel. The heavier constituents trickle down the plates, against the now retarded airstream, until they find a part of the plate sufficiently warm to vaporize them. The temperature of the chamber can be controlled so as to prevent superheating of the mixture and consequent expansion. The mixture is practically uniform and dry as it leaves the chamber. The very slight restriction introduced by the addition of the vaporizer is much more than compensated for by the effect of the greatly enlarged manifold passage made possible by the dryness of the mixture. Since no drops of liquid fuel are held in suspension in the airstream, the velocity of this stream can be kept at a low value.

Power and Economy Improved

Numerous lantern slides of graphs were exhibited to show the improved performance in power and economy of different types of engine with which the authors conducted the experiments, and in conclusion it was specifically stated that pre-manifold vaporization should offer some relief from the difficulties encountered in the distribution in eight-cylinder engines. The authors asserted also that better results could be obtained with dry mixtures such as are produced by effective vaporizers if the inlet-valve ports and valve areas were considerably enlarged.

The discussion included numerous questions and answers relating to further details of the experiments, the vaporizing device and methods in general.

At the Jan. 7 meeting of the Section, A. C. Mecklenburg, vice-president and chief engineer of the Murdock Pump Co., of Indianapolis, is to present a paper entitled, Advantages of Supercharging Internal-Combustion Engines. Mr. Mecklenburg is one of the pioneers in the design and development of superchargers, and it is understood that several manufacturers of high-powered cars are about to announce the equipping of their engines with the superchargers he will describe.

Front-Drive Experience Reported

Muller's Paper at Cleveland Section Brings Out Timken Test Results and Lively Discussion

CLEVELANDERS to the number of more than 200 appeared at the Dec. Section meeting at the Hotel Cleveland to hear and discuss William J. Muller's paper on Front-Wheel Drives, and were treated to authoritative discussion by Section-Chairman W. E. England, who had given the subject much study in connection with Stearns design, and Ernest Wooler, who reported on the experience gained in about 6000 miles, driving by himself and other engineers of the Timken Roller Bearing Co.

Dinner and entertainment preceded the regular session, which opened with the election of Orrel A. Parker as representation on the Sections Committee of the Society, J. Webb Saffold as Section representative on the Nominating Committee of the Society, and E. M. Schultheis as Mr. Saffold's alternate.

The name of Webb Saffold was heard again when the Chairman, W. E. England, called upon him to preside at the technical session, for which he had made the arrangements. Mr. Muller then read the paper on Front-Wheel Drives, which he presented at the Metropolitan Section meeting in November as reported in the S. A. E. JOURNAL for December, p. 696. The paper has to do particularly with the problems

met in the design of the Ruxton car by the New Era Motors, Inc., of which Mr. Muller is vice-president.

Some Design Difficulties

Chairman England, speaking from the floor, was the first discusser. He commented favorably upon the freedom given to designers of bodies for front-drive cars, the only limitation being that the driver must not be placed so low as to need a periscope. However, he considers as serious difficulties the lack of sufficient room for radiator area and the increased length of hood and wheelbase that are natural if not inevitable.

In the Stearns cars, designed by Mr. England, the engine completely fills the hood, he said, and the radiator is just sufficient for the worst conditions met in the United States. The whole car could be lowered 1½ in. by reducing the thickness of the rear cushion at the center, at the expense of a less comfortable ride for the middle passenger in the rear seat. The front seat can be lowered more than that amount without impairing the driver's vision. A layout was made to determine the results of charging this to a front-drive car, and it was found that 4 in. would be taken from the bottom of the radiator. This study showed that a car could be made 2½ in. lower by adding 9 in. to its wheelbase.

Observations from Driving Experience

Written discussion from Mr. Wooler, chief engineer of the Timken Roller Bearing Co., was read by Chairman Saffold. Mr. Wooler reported that he had driven a front-drive car about 2000 miles and that other engineers of the company had run the total mileage up to about 6000 during the last two months, to gain experience with the construction and the bearing problems. First impressions were very good. The car gave a new thrill because of the high speeds and feeling of safety on curves. A speedometer reading of 97 m.p.h. was obtained and a trip was made from Toronto to Cleveland in heavy rain, the distance of 325 miles being covered in less than seven hours.

No trouble was experienced from tire wear, wheel wabble, shimmy or other bad results of questionable features of design. No skidding was observable until heavy snow came on top of icy roads. A tendency to skid was then noticed when the car was going at a speed of 50 m.p.h. and the accelerator was released.

A front-end smash-up resulted from driving the car into a brick post. The front axle and brake-drum were bent, and the aluminum casting supporting the brake-shoe was broken. The damage was not thought to be more expensive to repair than would have been that needed by the radiator and axle of a conventional car.

Steering qualities and front-seat riding-qualities were reported as very good, but only fair riding in the rear seat. Three characteristics that might not occur to anyone without actual driving experience with such a car are: (*a*) Very high side-winds do not affect the steering. (*b*) The problem of battery location is serious. The position directly behind the fan makes the battery so hot that a fingernail will make a depression in its pitch insulation. The battery also has to be of special design, with baffles to prevent throwing acid over the engine. (*c*) The front-drive car is much dirtier than the rear-drive car because the dirt thrown up by the front wheels strikes the rear window light and other parts of the car.

Mr. Wooler's comments were confirmed for the most part by Paul C. Ackerman, assistant chief engineer of the Timken company, who nevertheless reported that he found no tendency to skid either on releasing the accelerator or applying the brakes when he tried out the car on ice.

Some Advantages Cited

General discussion was very lively, eliciting statements from Mr. Muller that tire chains seldom are needed and that they do not affect the steering when used; that one Ruxton car has been driven more than 10,000 miles without wearing the tread pattern off the tires; that a car has been dismantled enough to change a clutch in 1½ hr.; and that he believes the front drive is inherently no noisier than the rear-drive design. When the first Ruxton car was assembled, considerable noise came from the front end of the transmission. This was greatly reduced by adding a cross-member at the front, with rubber insulation.

Other discussers emphasized the fact that conditions as to riding comfort and noise will undoubtedly be improved in front-drive cars after they have undergone a longer period of development. Mr. England cited an experience with an engine which was very quiet on one chassis yet proved very noisy when applied to a different chassis.

Chairman Saffold quoted a remark overheard in the lobby of a bank where a front-drive car was on display, to the effect that it was a very beautiful car but would never be popular with bootleggers because it would not carry enough pay-load. Mr. Saffold thinks too much room is taken up by the "clock-work." He also mentioned as significant the observation that toy-shops sold far fewer toy wheelbarrows than formerly; it is more normal to pull a cart than to push a wheelbarrow.

Heldt on Front-Wheel Drives

New England Section Hears Well-Known Engineering Editor on Problems of Design

BONES of *The Horseless Age* were rattled at the meeting of the New England Section on Dec. 11, when Chairman W. M. Clark introduced P. M. Heldt as the man who made that publication an authority for nearly two decades. The meeting was held in the crystal room of the Hotel Kenmore in Boston and the attendance was a little more than 150.

The paper by P. M. Heldt, now engineering editor of *Automotive Industries*, was upon the general problems of front-drive design and did not include complete descriptions of cars, but following presentation of the paper Mr. Heldt presented a series of slides in which structural details of both the Cord and the Ruxton cars were shown.

Designers have resorted to almost every possible expedient in an endeavor to lower the height of cars. Some European sport cars are made with a tunnel in the floor to provide room for the propeller-shaft, but no such design has been found acceptable in this Country, as the mutilated floor is objectionable in appearance and inconvenient.

The advent of front-wheel driving for passenger-cars has come during a period when the eight-cylinder in-line engine is popular and four-speed transmissions are gaining in favor. Mr. Heldt called attention to the fact that this condition greatly increases the difficulty confronting the designer of a front-drive car, because one of his greatest problems is to keep down the length of the hood that is required to cover the powerplant. Abnormally long wheelbases are undesirable, both because of the additional weight and because they reduce the maneuverability of the vehicle.

Possibility of Improving the Ride

Emphasis was placed on the improved riding-qualities possible with a front-drive car because of reduction in unsprung weight; and an explanation was offered for the better road-holding ability of such cars on turns. Problems of mechanical detail connected with the universal-joints and steering-knuckles were given consideration, as was also the provision for cooling the engine in the reduced space left at the conventional position of the radiator.

Diagrams were shown to make clear

the advantage of moving the flywheel from the central position occupied in the conventional car and to illustrate the shifting of weight from the front to the rear axle that occurs during acceleration and hill climbing. Other slides showed details of the latest design of the British Alvis front-drive car. Not finding satisfactory an arrangement in which the two engines were geared together at the rear and transmitted power to the front axle through a central shaft, the Alvis now has a transmission placed in front of the axle. The transmission is substantially cylindrical, and no direct drive is provided. Brakes are located on both sides of the central housing, and the front wheels are independently sprung.

One common feature of European front-drive cars reported by Mr. Heldt is independent springing of the front wheels. This has not been adopted on the American cars, which have front axles bent to clear the gear-housing.

Veterans Contribute Experiences

Day Baker reported experience with industrial trucks used a number of years ago for climbing the gangplanks of ships. He found that front drives would climb much better than rear drives under these conditions. He also said that he had once been unable to penetrate a snow-drift in the ordinary way with a conventional rear-drive car, but had turned around and backed through it successfully. The tire chains ate their way through the snow-bank and made tracks in which the following front wheels ran easily.

Interesting tests on a four-wheel-drive electric truck were recounted by H. B. Church. The front and rear motors were disconnected in turn, and the current required for driving the car was measured by an ammeter. No difference was found on a straight, level road, but much less current was consumed while propelling the car in a circle, with the front wheels deflected to an extreme angle, when driving with the front motors.

Questions brought from Mr. Heldt statements that very little power is lost in a universal-joint and that its durability depends chiefly on its size in relation to the work to be done. He warned against throttling the engine in a front-drive car during a turn without releasing the clutch, because that might cause a skid; but he is convinced that slipping the front wheels is not so dangerous as slipping the rear wheels.

The meeting closed with a film showing the action of the Relay rear axle for motor-trucks, after L. A. Graham, vice-president of the Relay Motors Corp., had explained at length the construction, in which the load swings below the center of the axle, and recounted the advantages claimed for it.

Nitriding Investigations

Research Metallurgist of Nitralloy Manufacturer Reports to Milwaukee Section on Tests

ROBERT SERGESON, of the Central Alloy Steel Corp., who was the speaker at the meeting of the Milwaukee Section on Dec. 4, gave a résumé of the development of nitralloy and the nitriding process, compared the results obtained by nitriding with those obtained by carburizing, and reported on the effects of various temperatures during the nitriding process.

Between the usual dinner and entertainment and Mr. Sergeson's address, representatives of the Section were elected for two committees of the National Society. W. A. Starck was elected to the Nominating Committee, with A. W. Pope, Jr., as alternate; and Fred M. Young to the Sections Committee, with Mr. Starck as alternate.

Alloy Steels Suited to Nitriding

Nitralloy was introduced into America in 1927 through the agency of a French company which had bought the Krupp patents outside of Germany. All steels for nitriding had been made in electric furnaces up to that time; the first produced by the open-hearth process was made in 1928 at the plant of the Central Alloy Steel Corp. in Massillon, Ohio. As the process has proved successful, open-hearth heats of 65 tons are now being run. Chromium-aluminum steels were used at first. The addition of nickel proved unsuccessful because the nitriding temperature is within the brittle range of chromium-nickel steels. Alloying with molybdenum produces tougher steel than the straight chromium-aluminum steels.

The first nitralloy made in the United States had contents of both chromium and aluminum ranging between 1.80 and 1.30 per cent for each. These ranges have been reduced to 0.80 to 1.30 for chromium and to 0.60 to 1.20 for aluminum. This makes a steel of approximately the same analysis as No. 4130 S.A.E. steel, with the addition of the aluminum; and the lowered contents of alloys, according to Mr. Sergeson, have aided considerably in the making, pouring, rolling, forging and machining of the steel, with no important loss in the nitriding or physical qualities.

Furnaces, Methods and Results

Slides were presented to show that the hardness of the nitrided steels is much greater than that of carburized and tempered steel to a depth of about 0.30 in., and other slides showed the effect of variations in the nitriding temperature.

Furnaces of various sorts for nitriding were illustrated with sectional drawings and photographs of a furnace having a pot 22 ft. x 34 in. x 30 in. A fan for circulating the ammonia atmosphere seems to be essential for uniform results.

Tinning and nickel-plating are the only known methods of preventing the nitriding from affecting portions of a piece that is being treated. Simultaneously increasing the temperature of the furnace and the flow of ammonia vapor causes the nitrogen to be absorbed into the steel at a higher rate; but close regulation is necessary, as work may be spoiled by too high a temperature.

Plain carbon steels cannot be hardened to any appreciable extent by nitriding, according to Mr. Sergeson, but chromium-vanadium and chromium-nickel steels respond to the treatment, the former reaching a hardness of about 600 Brinell in 90 hr. Nitralloy decarbonizes badly at about 1600 deg. fahr.

Nitriding Increases Strength of Steel

Tensile tests of steel that has been heat-treated before nitriding show that the nitriding does not make it brittle nor introduce internal strains as does carburizing and quenching. Fatigue tests are said to show a considerable increase in fatigue resistance in the nitrided case. Samples heat-treated to an elastic limit of 73,000 lb. per sq. in., when tested in a fatigue machine under a stress of 75,000 lb. per sq. in., failed after 40,000 reversals. After nitriding, said Mr. Sergeson, the elastic limit was raised to 83,000 lb. per sq. in. and a sample ran to 28,000,000 reversals. Raising the stress to 85,000 lb. per sq. in., the piece failed after 195,000 reversals.

Mr. Sergeson said that the greatest industrial application of nitralloy so far has been in valves and valve-seats for superheated steam and re-

frigerators and for gears. Nitralloy is working very successfully as automotive piston-pins, pump-shafts and crankshafts.

Questions elicited further information from Mr. Sergeson, including the statement that nitralloy should be ground with a soft wheel to avoid excessive heat, which may cause cracking of the surface similar to that found in a carburized case. It is slightly more difficult to machine than corresponding steels without the aluminum inclusion. Resistance to abrasion is estimated as about five times that of case-hardened steel.

Nitralloy cannot be made quite so clean as other grades of steel, because of aluminum oxides, but this has caused no difficulty in the way of soft spots. Annealing nitralloy is not practicable, although it can be done in hydrogen. Large parts require 80 to 90 hr. for nitriding and automobile pump-shafts about 30 hr. J. C. Slonneger, of the International Harvester Co., reported a case of about 0.015 in. depth having been obtained on pump-shafts in that time by the nitriding process.

The Milwaukee Section will hold no meeting during January, Chairman Arthur C. Wollensak announced, because so many of the members will be absent during the month at the various automobile shows.

Factors Affecting Truck Efficiency

Pittsburgh Section Analyzes Economic Considerations as Outlined by J. F. Winchester

DIVIDING his subject into highway, transport and vehicle efficiency, Mr. Winchester, general superintendent of automotive equipment for the Standard Oil Co. of New Jersey, subdivided these divisions and outlined, with the assistance of lantern-slide views, their various relationships in the paper he presented at the meeting of the Pittsburgh Section held at the William Penn Hotel in Pittsburgh on Dec. 6.

The other paper was by K. A. Dalsky, assistant manager of the motor-truck tire department of the Goodyear Tire & Rubber Co. of Akron, Ohio, who first offered a definition of a tire as "a machine built of rubber, fabric and steel, designed to compensate by movement within itself for the irregularities of the road surface over which the vehicle passes and thereby keep the axle of that vehicle on as nearly a uniform horizontal plane as is possible." If this definition be an accurate one, said Mr. Dalsky, we must consider the value of different types of tire from a study of their relation to the vehicle and to the work the vehicle must do.

The meeting was preceded by a dinner and musical entertainment attended by 99 members and guests, and when Chairman Nils G. Bjorck convened the technical session at 8 p.m., the total attendance was 135. After a brief business session relating to nominations for and election of members who will represent the Section on the Nominating Committee of the Society, which was reported upon by Prof. James W. Trimmer, the meeting was turned over by Chairman Bjorck to J. M. Orr, general manager of the Equitable Auto Co., Pittsburgh, who introduced Mr. Winchester.

While exhibiting his numerous lantern-slides and commenting upon them, Mr. Winchester remarked that, in his opinion, we are on the verge of great development in the design of six-wheel trucks and that, within the next few years, their number will be greater than the number of four-wheel trucks operated. He based this belief on his statement that a larger pay-load can be carried with smaller gross weight involved on the six-wheel than on the four-wheel vehicle, and that statistics show that the six-wheel vehicle will not be so costly to taxpayers as is the four-wheel type. He also showed views of several types of tractor-trailer motor-vehicle equipment used for special purposes and vehicles specially designed to serve different special purposes, commenting on the advantages and disadvantages of each. He also exhibited views of special tools and shop equipment, and described the methods used by his company in their application to repair and reconditioning jobs.

In the interesting discussion which followed, numerous points regarding large-scale fleet-maintenance were brought out, among them being cost-keeping methods, the use of differentials between the rear axles of six-wheel units, various details of dynamometer and other tests, and the results obtained from the use of air brakes, with which virtually all the vehicles supervised by Mr. Winchester are equipped.

Study of Riding Comfort

Indiana Section Meeting on Principles and Accessories That Affect the Ride

WINNING at football is a novel ground for the selection of an S.A.E. speaker, but the record of the Purdue team was cited as one of the reasons for asking Prof. H. M. Jacklin to present the leading paper at the meeting of the Indiana Section at the Hotel Severin on Dec. 12. The meeting was carefully planned to cover riding-qualities in general and springs and shock-absorbers in particular. It was preceded by a dinner in the Roof Garden and opened with the election of Louis Schwitzer as a member and H. C. Snow as alternate for the Nominating Committee of the National Society, and of Frederick S. Duesenberg as a member and George H. Freers as alternate for the Sections Committee of the Society.

Chairman Bert Dingley then introduced Professor Jacklin, who presented his paper on Riding-Qualities and Their Measurement, in which he stated the complex factors involved in judging riding-qualities, reported how various experimenters have tried to establish numerical values by which to rate vehicles or equipment, and explained the status of research work at Purdue on riding-qualities.

For definition, Professor Jacklin said that an automobile has good riding-qualities when it transports its passengers in such a way that the trip is a pleasure and not a hardship. Through the ages, the human being has become accustomed to rates of movement within a range of 70 to 140 complete vibrations per minute. Marching cadence is 120 vibrations per minute. The most comfortable cars have suspensions that vibrate from 100 to 140 in front and from 70 to 90 in the rear. Pitching results when the rates at the front and the rear are too nearly alike, and reducing the number of vibrations below 70 per min. causes trouble from having the wheels leave the road for too long a time.

Discomfort from Horizontal Thrusts

Diagrams were presented to show how softer tires result in smaller hori-

zontal thrusts, which Professor Jacklin said are an important cause of discomfort, and how larger wheels would contribute to comfort. Increasing the wheelbase of a vehicle decreases its period of vibration, so it does not necessarily add to the comfort of the vehicle. The importance of lower unsprung weight was also emphasized.

Since 1928, when Purdue was asked by the Society to participate in riding-quality research, much study has been given to the problem but little money has been available for instrumentation. The proposed program includes the procuring and developing of instruments and methods for measuring and recording forces and their rates of change and personal-comfort reactions, and the correlation of the opinions of many subjects. There probably is an opportunity for some man to earn an advanced degree in prosecuting this work.

Springs and Shock-Absorbers

Moderation added force to the contribution of J. M. Schoonmaker, Jr., vice-president of the Standard Steel Spring Co., who admitted that most of the great general improvement in the riding-quality of cars is due to improvements in shock-absorbers and tires. He confirmed the conclusion of other engineers that semi-elliptic springs are chosen mainly because of engineering limitations rather than because one type of spring gives better riding-qualities than another, and said that the rear of the car should have as long a period of oscillation as is practicable. Limiting factors are the undesirable appearance of a car in which provision is made for an abnormal amount of rear-spring deflection and the excessive chatter that results from an extremely low rate. General use of shock-absorbers has made possible the use of springs having lower rates.

Papers on shock-absorbing devices were read by Frank C. Born, chief engineer of the automotive division of the Wahl Co., of Chicago; and F. L. Mills, of the engineering department of the Gabriel Snubber Mfg. Co., of Cleveland. Mr. Mills divided shock-absorbers into five classes and described the action of each class, adding a description of the Gabriel triple-hydraulic shock-absorber. This is a double-acting device with drop-forged steel reservoir and vane. He closed with a plea to designers of automobiles to make provision for the mounting of shock-absorbers before the car design is advanced so far that a suitable mounting is difficult to arrange.

Fred Duesenberg presented a communication in which John Warren Watson stated that better riding-qualities will depend upon finding a way to provide more amplitude for spring motion. He believes that almost all of the discomforts now suffered at the rear seat are due to excessive stiffness of the front springs resulting in high period angular movements of the rear seat. Mr. Duesenberg said that Mr. Watson once claimed to know more about springing an automobile than he did about shock-absorbers, and that he has said repeatedly, "Your springs are too stiff; you can't get a good ride that way."

Remarks on the subject of shock-absorbers were made also by J. R. Rix, of the American company which is making the Baby Austin car in Butler, Pa., and S. J. Swift, of the Cleveland Pneumatic Tool Co.

Members of the Section had been invited by the Curtiss-Wright Flying Service to witness the arrival of the night airmail and other special activities at Stout Field, but the fog was so bad that Captain Barnes, who attended the meeting, said there was zero ceiling and zero visibility and no mail was in the air. He invited automotive engineers to attend one of a series of eight lectures the Curtiss service is giving. This might be termed a survey course, in which the principles of aviation are given and illustrated by specially arranged flights that are complementary to the lectures. The cost of the course is $75, including 3½ hr. of diversified flying.

Chairman Dingley announced that the Indiana Section will hold no meeting in January and that the February meeting will consist of reports on the new mechanical features of cars displayed at the various automobile shows.

Transmissions and Diesel Engines

Demonstrations, Motion Pictures and Slides Show Latest Practice to Northwest-Section Members

POSSIBILITIES with regard to subdividing the present Northwest Section into two Sections, one being in Seattle and the other in Portland, were discussed under the Chairmanship of Robert S. Taylor at the meeting of the Section held Dec. 14 at the Gowman Hotel in Seattle; but the matter was not decided and will be studied further before definite action is taken. The Section now has members in both Seattle and Portland, but the 200 or more miles separating these cities is making it difficult for members to attend.

Among the interesting features of the technical session were a description of the Reo transmission, by George Nyberg, who used an actual transmission in describing its design and operation. The transmission was said to have been invented by Messrs. Fertrand and Thomas of the Reo company's factory about four years ago, and to have been developed through experimentation since then to its present design.

Four-Speed Advantages Depicted

Cornelius E. Tierney, of the Graham-Paige Co., exhibited interesting motion pictures of his company's four-speed transmissions which showed the advantages of this type of drive. Mr. Tierney said that this arrangement provides a practical and fast rear-axle ratio, slows down both engine and propeller-shaft speeds, tends toward economy of gasoline and oil, lessens vibration troubles and runs quietly. He stated also that it increases the length of life of the engine and all related moving parts, provides better cooling and affords greater comfort for the occupants of the car.

Mr. Tierney compared two cars; one having four speeds, the other, three. He considered a 4000-lb. three-speed car equipped by its manufacturer with a 4.88:1 axle-ratio which was regarded as giving excellent performance with that ratio. At a car speed of 65 m.p.h., the engine speed is 3500 r.p.m., which is considerably faster than the speed at the peak of the horsepower curve. With a 3.69: 1 ratio as used with four-speed transmission, the engine speed at 65 m.p.h., was only 2700 r.p.m. and the peak of the horsepower curve had not then been reached, while the torque curve dropped only a small amount. Motion-picture films were shown to illustrate the details of this type of transmission, and its advantages were enumerated.

Diesel Engine for Automotive Uses

Showing numerous lantern-slide views of different types of Diesel engine illustrative of the points brought out in his paper, Prof. F. G. Baender, of the Oregon State Agricultural College, said in part that the characteristics of the Otto-cycle and the Diesel types of engine are as follows; stating also that the Diesel engine approaches more nearly the action of the steam engine in that it will show wonderful pulling power at low speeds:

OTTO TYPE

Adiabatic compression
Adiabatic expansion
Mixture burns or explodes at constant volume
Needs external source of ignition
Needs carbureter or premixing
Variable compression
Very poor torque at low speeds
High-temperature exhaust
Exhaust contains carbon-monoxide gas

DIESEL TYPE

Adiabatic compression
Adiabatic expansion
Mixture burns at constant pressure
Fuel ignites from heat of compressed air
Needs no carbureter
Constant compression
Good torque at low speeds
Not so high a temperature of exhaust
Practically free from carbon-monoxide gas

Professor Baender then gave an elementary résumé of the fundamentals of the theory of heat, his reason being that an internal-combustion engine is essentially a heat engine. He outlined the combustion problems involved in the Diesel engine and compared them with those of the Otto-cycle engine.

Mentioning the two types of Diesel-engine classification, two-cycle and four-cycle, the speaker said that so far as he knows all Diesel automotive engines in this Country are of the four-cycle type. The manufacturers of two-cycle engines claim for them the elimination of all complicated mechanically actuated valves, a more simple cylinder-head construction, no rocker-arms, push rods and timing gears, fewer moving parts, operating simplicity, no valves to grind and twice as many power impulses in the same number of revolutions. The advantages of the four-cycle type claimed by the manufacturers are that these engines are more economical, burn heavier fuel better, allow the bearings to be kept in order more easily as the loads are intermittently relieved, the temperature stresses are less severe, the cylinder castings are less complicated, greater reliance can be placed on the engine, and that the expense for upkeep and major repairs is less. Operating characteristics of the Diesel engine were outlined by Professor Baender, who also discussed Diesel-engine fuels, lubricating oils, operating troubles, and operating-cost data.

Discussion of the paper consisted mainly of queries and answers giving further details and explanations of the points brought out by the speaker.

N. Y. U. Student Branch

A NEW student branch of the Society was officially inaugurated at New York University on Dec. 18 by President-Elect Edward P. Warner and Research Manager C. B. Veal. More than 40 attended the meeting, which was convened by Chairman Hem, of the branch, who introduced Mr. Warner and Mr. Veal. Plans are to be made this month by the officers of the branch for a program of activities for the remainder of the season.

In his remarks, Mr. Warner said in part that officers of the Society get in touch at Student Branch meetings with men who will become active in the industry in a few years and the students come into contact with those who are already established in it. The students should make the most of the opportunity afforded by their Branch activities. It is well to have faculty advisers, but they should not become controllers so that the branch becomes a glorified part of the university curriculum.

The Society wants the students to make use of the experience they have acquired, no matter in what field it has occurred, for, although originally interested primarily in automobiles, the Society has extended its activities both vertically and horizontally to include motor-trucks, motorcoaches, motorboats and airplanes from the raw material to the finished products and their use. To think that we derive all our information from factories and laboratories is a mistake. The kind of knowledge gained from trying to persuade a ramshackle old car to go from here to there is as useful as more detailed facts obtained from factories.

The activities of the Society were expanded because the interdependence of the various branches of the automotive industry were realized. An effort is made to overcome specialization of knowledge by acquainting each part of the industry with what the other parts are doing, so that progress will be at the rate of the fastest.

Mr. Warner expressed the hope that the N. Y. U. Student Branch will be kept from becoming merely a place in which to read papers; it should have the social aspect and, most important, should do some research work. This could be along lines paralleling those of the Society; for example, with faculty cooperation a committee on standards could be set up to function. One line of research that he should like to see taken up is in the field of distribution, since about 30 per cent of the price of an article represents distribution costs. In the proper handling of this problem may lie the solution of business cycles of great activity succeeded by depression.

Mr. Veal, speaking to the students, urged the value of debates in keeping up interest and extending knowledge of the subjects. He recommended cooperation among Student Branches not too remote from one another.

Section Debate on Six-Wheelers

Southern Californians Disagree on Two-Wheel and Four-Wheel Rear Drives—Warner Talks on Separatism

SOLAR energy for which the region is so much vaunted seems to be released in large quantities by attendants at the meetings of the Southern California Section, which always are characterized by enthusiasm, keen interest and animated discussion. The meeting at the Los Angeles City Club on Dec. 6 was a good example of the kind of meetings the Section puts on. Much of the success of the meetings doubtless results from the fact that operators and maintenance men always are present and do not hesitate to criticize, from their background of experience, the theories and claims of the design engineers and constructors. Thus, at the December meeting, users disagreed with the assertions of engineers regarding two-wheel-drive six-wheel vehicles. Three papers on six-wheel trucks were presented, one by Edward Dagner, factory representative of the Fageol Motor Co.; one by F. B. Tucker, president of Six-Wheels, Inc.; and one by Ethelbert Favary, consulting engineer of the Moreland Motor Truck Co. In the discussion, Col. William H. Fairbanks, supervisor of shops and vehicles of the Southern California Telephone Co., and several others told some of their experience in the use of six-wheel vehicles.

Two events out of the usual were the presence of and short addresses by Edward P. Warner, President-Elect of the Society, and Edward C. Wood, of the Northern California Section. Vice-Chairman F. C. Patton presided during the meeting in the absence of Chairman Fairbanks, who was delayed in returning from San Diego but arrived later.

Warner Warns Against Sectionalism

Upon being introduced, Mr. Warner spoke energetically on the topic of National solidarity of the S.A.E. and the desirability of maintaining the Nation-wide integrity of the organization. While admiring the noticeable tendency of the Far West to pride itself upon

being self-dependent, he said he thought we should be on guard against a feeling that the Country is too large to be covered as a unit by one professional organization. It is particularly appropriate that the Society should set an example of being a fully National and coordinated organization, because there is less tendency to geographical separatism in the automotive industry than in any other even approaching it in magnitude.

Mr. Warner mentioned briefly the changes in the organization and extension of its activities that have made it truly National in scope and comprehensively inclusive in the automotive field, embracing as it does the production and operating engineers as well as designers. The work that is being done in the western Sections, as embodied in papers and reports in the S.A.E. JOURNAL, he said, is of pronounced service to members in the East and he hopes that the work of the Society and its eastern Sections, as embodied in THE JOURNAL and TRANSACTIONS, is of great value in the West. He spoke of the vital importance of the function of the Society in bringing engineers together for mutual discussion of their common problems, and of the social advantages of the Section meetings, and said that the Society will not be performing its full work properly unless the organization is used to bring about a better understanding of one another's problems by those who design, manufacture, distribute and operate motor-vehicles and airplanes.

Mr. Wood brought greetings from the Northern California Section and a suggestion that the Southern and Northern California Sections endeavor to have a joint meeting at some half-way place for the exchange of operating experiences in the two parts of the State.

Six Reasons for Four-Wheel Drive

Mr. Dagner gave six reasons, with explanations, for the use of four-wheel-drive six-wheel trucks. These are (*a*) lower cost per ton-mile because of increased pay-load, (*b*) reduction of impact on the highway, (*c*) increased tire life, (*d*) easier riding for load and driver, (*e*) equalized four-wheel braking by the use of air-brakes with individual diaphragms for the four drive wheels, and (*f*) full conformity with State laws through accurate weight distribution of the chassis and the load.

Foundation was laid early in the paper for the later controversy by the statement that, although attachments consisting of a non-driving load-carrying auxiliary axle designed for the purpose of complying with the letter of the law have met with a very ready market, their use merely tends to decrease the factor of safety built into the standard components of every motor-truck and that the apparent economies in cost per ton-mile are not real, as the element of time or the life of the equipment is overlooked. He also commented on the importance of safety to the public as affected by increasing the load beyond the capacity of the steering and braking mechanism as built by the manufacturers for two-wheel-drive four-wheel trucks.

Six-Wheel Attachment Defended

Mr. Tucker, in his paper, upheld the idle load-carrying third-axle attachment on the basis of saving in the total cost per ton-mile of hauling, including initial cost, operating expense, maintenance, repairs and insurance. He divided six-wheel trucks into those which seldom operate on modern highways and the much larger class that operate almost entirely on modern highways. He maintained that the four-wheel-drive types suitable for the former class of service are expensive to build and uneconomical for the second class of service. He predicted that trucks for the latter service will increase in number as the mileage of good highways increases, and said that the two-wheel-drive six-wheel truck unit, designed to be incorporated in the vehicle either at the time of its assembly or after its delivery to the operator, was brought out to enable the operators in this larger field to reap the financial advantages to be derived from improved roads. He asserted that the two-wheel-drive far outnumbered the four-wheel drive six-wheel vehicles because of the lower initial cost, larger pay-load percentage and smaller percentage of operating cost chargeable to tare weight, and pointed out that not very long ago it was not uncommon to see a truck weighing as much as 8 tons loaded to maximum legal gross weight with only 4 tons of pay-load. As for traction, he said that thousands of two-wheel-drive six-wheelers are regularly negotiating mountain grades under all weather conditions and in some cases pulling trailers, the total gross weight of truck, trailer and load aggregating 34,000 lb.

Four-Wheel Drive Development

Mr. Favary's paper referred to the six-wheel trucks described in a paper he presented nearly five years ago at the first meeting of the Southern California Section which were the only four-wheel drive six-wheelers then in production in this Country, so far as he is aware. They were called 10-ton trucks and of 34,000-lb. gross weight with load. After observing the operation of many hundred six-wheel trucks for several years, he said, it can be definitely asserted that the cost per ton-mile is considerably lower with the six-wheeler than with the conventional four-wheel vehicle. He described and illustrated some of the mechanical refinements made since the earlier six-wheelers were designed, which have resulted in easier riding, lower unsprung weight, reduced road impact, and increased tire life. He referred to the weight of the spring in the early truck as 375 lb., which was questioned later in the discussion but was verified. In the later construction the springs are about 100 lb. lighter and their weight is carried on the sprung part of the chassis. The total saving in unsprung weight by modifications made is more than 400 lb. on each side, or 800 lb. at the rear end of the truck.

The Moreland company now manufactures four other sizes in addition to the 10-ton model, said Mr. Favary, the smallest being of 16,000-lb. gross weight with load.

Main Points in Discussion

Some written discussion on the angle of universal joints in six-wheelers, illustrated with lantern slides, was presented by John Wiggers, of the Moreland Truck Co.

Harold E. Larson, of the Timken-Detroit Axle Co., said that, as axle builders, his company believes that in the future the orders for two-wheel-drive four-wheel trucks will be as few as for the solid-tire truck today. Whether a third differential in a four-wheel-drive six-wheeler will warrant the extra cost and weight is uncertain, but operation by the factory in Detroit of an experimental truck with such a differential has shown that the differential has increased the tire life. Since last February a four-wheel-drive six-wheel truck hauling a gross weight of 34,000 lb. and a 34,000-lb. trailer load has run 50,000 miles on its original set of tires. It has an ordinary two-differential axle-unit. He said that the company cannot guarantee the usual two-wheel-drive axle that has a capacity of 22,000-lb. gross vehicle load if the six-wheel attachment is put on and the same truck and units carry a 34,000-lb. load, for surely the extra idle axle does not carry the 12,000-lb. excess.

Referring to the weight of springs, John B. Ronen, of the United States Spring & Bumper Co., verified Mr. Favary's figure of 375 lb. and said that not more than a month previously his company made a spring having a total weight of 420 lb. with its auxiliary springs. Change from the single to the double type of spring with an auxiliary spring, as incorporated in the Moreland trucks, eliminated much weight and gave the same carrying capacity.

E. M. McLean, West Coast representative for F. W. D. trucks, mentioned the company's six-wheel-drive construction and said that it will soon have on the road a 34,000-lb. truck of this type, which drives on the front two as well as the four rear wheels and has a center differential.

Effect of Weight on Traction

To illustrate the importance of carrying as much of the weight as practicable on the drive wheels so as to get maximum traction, Mr. Favary told of a farmer who was driving a donkey hauling a load of potatoes. When they came

to a hill and a slippery road, the donkey slipped and could not draw the load up. So the farmer, after some thought, took two sacks of potatoes from the cart and put them on the back of the donkey, which then went up without slipping. In engineering language about coefficients of friction, about which the farmer knew nothing, Mr. Favary vigorously pointed out the fallacy of believing that, unless the weight on the driving wheels is increased, the attachment trucks will have any more tractive ability than the four-wheel trucks. He further maintained that only a certain amount of work can be got out of an engine and transmission and if they are over-worked they will wear out sooner. As for a third differential, where the drive wheels are close together, as in the four-wheel-drive six-wheeler, the pneumatic tires compensate sufficiently for the difference in turning radii to prevent excessive tire wear; but the Moreland company is investigating the fact that increased tire mileage is obtained with the third differential.

Colonel Fairbanks recounted in an interesting way some experiences with trucks used by a construction crew of the telephone company in hauling poles and wire for a line from San Bernardino, Calif., to Las Vegas, N. M. Moreland, Fageol, White, Mack and F.W.D. trucks were used. Most of the heavy hauling was done with the four-wheel-drive six-wheelers, which gave very little trouble, although the cylinders required regrinding after the trucks had run about 10,000 miles, because of the sand. Sometimes the drive wheels would dig into the sand so deeply that a Holt tractor was used to pull the trucks out. Excellent results were obtained with a Ford truck having a four-wheel-drive rear end built up from Ford parts. The trouble found with the F.W.D. six-wheeler driving on the front wheels and one pair of rear wheels was that it did not have enough traction in the sand. However, Colonel Fairbanks believes that the two-wheel-drive six-wheel truck fills a distinct field and that it has sufficient traction for operation on the highways. With the extra axle under the rear, he thinks the factor of safety built into the components is great enough to carry the heavier load, but sounded a note of warning about providing adequate brakes.

Tires Part of Transmission

Effect of Car-Design Changes on Tires and Brake-Lining Told at Chicago

PERHAPS a new conception of the functions of the pneumatic tire was given members of the Chicago Section at their monthly meeting on Dec. 10 in the College Inn of the Hotel Sherman when R. D. Shields, of the Firestone Tire & Rubber Co., told them that "the rubber tire on which the car rests is as much a fundamental and vital portion of the transmission mechanism of the car as is the clutch of the transmission gears or the axle. Every ounce of power that is developed by the engine must be passed through the fabric structure of the tires before the power can move the car a single inch."

The speaker, in introducing his subject, mentioned the reduction of the number of passenger-car tire sizes from about 45 a few years ago to 18 standard sizes at present and referred to the advent of balloon tires and the troubles they introduced for the car-design engineer. Experience proved that the four-ply tires introduced car instability because of the large cross-section and the low air-pressure of 20 to 25 lb., hence the industry gradually tended toward the adoption of the six-ply tire, which promises to come into almost universal use for the larger cars.

Mr. Shields' theme was the development of tires as related to the requirements of modern passenger-cars. Most persons do not realize just what effects recent changes in motor-cars have had on tires, he said. Frank admission was made that within the last year many more complaints have been made regarding rapid tread-wear than in all the history of the tire company. But, when the conditions that give rise to the complaints are analyzed, the reason for them can easily be seen; the answer is comprised in the two words: "increased power."

Power Increase Overlooked When Tiring

When the functions of the tire are considered, most persons think of the tire merely as a cushion to absorb the jolts and protect the mechanism of the chassis and the occupants of the car from vibrations. But the function that is just as important but almost universally neglected is that of transmitting the power of the engine. Whereas the horsepower of a high-class car some years ago was commonly from 50 to 60, it is now going up to 100 or more; Fred Duesenberg is now even advertising that his car has 240 hp.

When the design engineer starts to bring out a new car model, he provides for a larger engine; then he must have a slightly larger clutch, put in stronger gears and probably increase the size of the axle somewhat. But when it comes to equipping the car with tires, even smaller tires may be fitted. We all subconsciously recognize that the faster a car goes, the harder work is required of the engine and transmission gears, but we forget the tires. If the speed of a car is doubled, only one-quarter the tire mileage should be expected, according to Mr. Shields, for the work the tire does is inversely proportional to the square of the speed. When more powerful engines are used and the acceleration and normal speed of the car are increased, followed with more effective brakes and more frequent traffic stops, it is not a cause for wonder that tread wear is rapid.

Another factor has been the reduction in wheel diameter; every inch taken from the diameter takes 10 per cent from the life of the tire. Heretofore, in choosing tire sizes for passenger cars, the tire companies have done so almost exclusively on the basis of car weight, but Mr. Shields believes that in the future it will be necessary to consider also the power and the speed.

Balloons for Trucks and Coaches

Prediction was made with regard to tries for motor-trucks and motor-coaches that in a very short time balloon tires will be used almost exclusively. The truck operator who is very careful of his equipment cannot fail to realize that easier-riding tires add to its life. Truck manufacturers are now almost without exception cooperating with the tire companies with the object of providing tires of sufficient capacity and making provision to fit oversize tires for carrying any load recommended for the chassis or allowed by the law. Another important move in this connection that the tire companies hope to see made is the adoption of a method of rating trucks according to the maximum allowable gross weight and the provision of the correct tire equipment for carrying that weight. Some truck companies have already done this.

The name balloon, as applied to truck tires is a misnomer, as they are not balloon tires in any sense of the word as customarily used; they are simply medium-pressure tires. The pressure in the Firestone truck and coach tire ranges from about 50 lb. per sq. in. in the smaller sizes to 75 and 80 lb. in the largest size, which is a 12.75-in. tire.

Discussers raised a number of questions pertinent to the subject and brought out interesting and informative explanations from Mr. Shields.

Lubricated Molded Brake-Linings

The second paper of the evening, on The Trend in Present-Day Brake-Linings, was presented by T. L. Gatke, of the Gatke Brake Lining Co., who said as a preface that the words "brake-

lining" could be used instead of tires in the address that Mr. Shields had given, as there is little difference between the arguments used by Mr. Shields and those used by Mr. Gatke's company. Vehicle engineers seem willing enough to make generous designs, but the director of purchases limits the amount of money to be spent for tires, brakes or whatever it may be. He created laughter when he said that the vehicle manufacturer has to have tires for the operation of his vehicles but most people have gone without brakes for a good many years; it is only within the last six months that he has noticed that people are really willing to buy brakes.

Most of Mr. Gatke's talk was about the molded asbestos-graphite lining which his company first brought out about five years ago. This lining was lubricated with graphite and he was the laughing-stock of brake people who had worked on the brake and lining problem before him. However, the graphite lining, just as it was originally made, is still in service and six of the ten largest motorcoach fleets in the United States are using the lining, he said, and at least 75 per cent of the motor-vehicles in the Country are now using molded lining.

Mr. Gatke proceeded to explain the advantage of molded linings, which retain their uniformity with use because they do not dry out and no chemical reaction occurs through a long period. A truly molded lining, he asserted, is almost certainly proof against these actions. He also discussed quality, the ill effects of buying linings at the lowest obtainable prices and highest discounts, the importance of obtaining linings that give long service, and the necessity of having adequate braking surface for the speed and weight of the vehicle.

Questions regarding the effects of moisture and grease on brakes, brake-testing machines, brake adjustment, composition of brake-drums, coefficient of friction and related subjects were asked in the discussion and Mr. Gatke answered.

Life Membership Awards Proposed

Various suggestions made at the opening of the meeting that an award or awards of life membership in the Society be made by the Chicago Section for notable achievement in the automotive line were discussed at length. The outcome was that the meeting voted for the appointment of a committee of five members to prepare a questionnaire on the subject to be mailed to the entire membership of the Section and to make a report on the result at some future Section meeting.

Papers for William C. Naylor Memorial Award

IN ACCORDANCE with the announcement of the William C. Naylor Memorial Award competition, as made by the Aeronautic Division of the Detroit Section, attention is here directed for the third time in the S.A.E. JOURNAL to the rules for the award, which were published in full in the November number of THE JOURNAL on p. 559.

Papers entered for the award must be sent to the office of the Detroit Section, S.A.E., General Motors Building, Detroit, in time to be in the hands of the Committee of Judges not later than Jan. 15, 1930.

Any author who is a graduate of a recognized college offering undergraduate work in aeronautical engineering is eligible for the award.

The award is to be made each year to the author of the best paper relating to analytical investigation in aerodynamics, aircraft design or construction, or experimental research in fundamental problems of the airplane.

The subject may be original with the author or a continuation of some research work bringing to light new aspects of the original investigation, and may be a revision of the senior thesis presented by the author for his bachelor's degree in aeronautic engineering.

The award consists of a certificate and a Junior Membership in the Society (initiation fee and one year's dues), and is to be presented at the closing meeting of the Aeronautic Division of the Detroit Section each season.

Shock-Absorbers Discussed

RIDING-QUALITIES as improved by the use of shock-absorbers was the topic discussed in a paper presented at the Dec. 18 meeting of the Dayton Section at the Engineers Club. The address was given by W. F. Whiteman, who was assisted in the discussion by E. F. Rossman. Both men are shock-absorber engineers of the Delco Products Corp. and they dealt principally with the means of that corporation in attacking the problem of riding-comfort. Numerous section models of the corporation's hydraulic shock-absorbers were exhibited for inspection, and lantern slides of the construction and results obtained with them on cars were used to illustrate the paper. The action of the device was thoroughly explained, with the aid of slides of curves.

Inclement winter weather interfered with a good attendance of members. Those present, however, took part in animated discussion of hydraulic shock-absorbers, riding-comfort, independent springing of wheels, and allied subjects.

Wichita Aircrafters Discuss Welding

WELDING as Applied to Aircraft Construction was the title of a paper presented at the meeting held by the Wichita Aeronautic Section on Nov. 21 in Wichita, Kan. The speaker was A. K. Seeman, of the Linde Air Products Co., who dealt with structural materials, welded steel fuselages, improvements in welding practice, procedure control, the qualifications of welders, design and layout of welded joints and welding technique.

Much informative discussion followed the presentation of the paper, and Mr. Seeman was questioned on numerous points.

Mac Short presided and, in the absence of Chairman Mooney, of the Program Committee, told of some of the interesting papers to be given at coming meetings of the Section. Those who took part in the discussion on welding were Mr. Short, of the Stearman Aircraft Co.; Walter Burnam, of the Travel Air Mfg. Co.; and Messrs. Seeman, Pickens and Clark, of the Linde Air Products Co.

Christmas Get-Together

THE Northern California Section meeting on Dec. 12 was a Christmas get-together stag party exclusively for members and applicants. The attendance was about 75; had members been permitted to bring guests, it could have been more than 200, as innumerable requests to bring guests were made.

The table was attractively decorated with flowers for the occasion and caps were given to the members. Several pleasing entertainment numbers were rendered and an unusual feature was the roll-call in which each member was asked to introduce the man at his left. A general get-together to promote mutual acquaintance followed the dinner. Everyone proclaimed the meeting a very successful affair.

The next monthly meeting of the Section is to be held on Jan. 9 at the Engineers Club in San Francisco, with a dinner at 6:30 p.m. followed by the technical session at 8 o'clock. Ten-minute talks are to be given on carbureters, governors, valves and bearings by speakers who had not been selected at the time of the December meeting.

Hiram Lew Wallace

WORD has been received of the greatly deplored death of Hiram Lew Wallace, research engineer in agricultural engineering at the University of Nebraska Agricultural College, in Lincoln, on Dec. 6. He had been ill for two weeks, and was found to be suffering from stomach trouble and an inflamed appendix. After an operation complications set in which proved fatal two days later.

Mr. Wallace had been connected with the University of Nebraska since 1925 as research engineer in charge of tractor testing in accordance with the Nebraska State law and also as research engineer in agricultural engineering. While at the University, he conducted some interesting experiments, one which was called a Basic Study of Tractor-Lug Efficiency being made in conjunction with Fred Laub.

Mr. Wallace was born at Bolckow, Mo., in September, 1892. His education was interrupted by the World War, as he enlisted in the United States Army and was sent to Rosny, France. He went into action July 28, near Sennecy, France, and was gassed in the second battle of the Marne and sent to the hospital, but he returned to his company in September. After spending the winter with his company in Germany, he returned to the United States the following summer and was discharged at Presidio, Calif., in August, 1919.

Mr. Wallace attended the A. E. F. University at Beaune, France, during its existence. In 1920 he received his degree of Bachelor of Science from Iowa State College, and in 1922 received his Master of Science degree. During the years of 1920 and 1921, he was a tractor-testing engineer in the agricultural department of the University of Nebraska. He then left Lincoln and became an instructor in farm machinery and farm engines in the agricultural engineering department of the Iowa State College at Ames, Iowa. This position was held until 1925, when he returned to the University of Nebraska.

Mr. Wallace was actively interested in National organizations which dealt with his line of work, and as a junior member of the American Society of Agricultural Engineers presented several papers before that Society; one, given at a National meeting held in Madison, Wis., on Depreciation of Farm Machinery, and another, Recent Changes in Tractors, presented at the Farm Power Meeting in Chicago. Mr. Wallace was a member of the American Legion and of various clubs and lodges in and about Lincoln. He was active in university affairs, being a member of the annual engineering inspection committee, and, though not a member of the teaching staff, lectured regularly to freshmen engineers on orientation and to other classes on special occasions. He was elected to membership in the Society in October, 1928.

Stanford A. Smith

A WEEK subsequent to an operation for acute appendicitis, Stanford A. Smith, Detroit branch manager for the John Warren Watson Co., passed away on Nov. 8.

Mr. Smith was born in May, 1890, at Waltham, Mass., where he received his early schooling and was graduated from the Waltham Technical High School in 1908. While attending high school he worked afternoons as an apprentice machinist with the Waltham Watch Co., thus laying a sure foundation for the work he was to do.

From 1910 to 1920, Mr. Smith was employed successively by the Yale & Towne Mfg. Co., at Stamford, Conn., and the Link Belt Mfg. Co., of Indianapolis, as die maker; the Continental Motors Co., of Detroit, as jig and fixture inspector and tool demonstrator; Maxwell Motors, of New Castle, Ind., as tool demonstrator and general foreman; the Timken-Detroit Axle Co. as assistant chief inspector at the West Side plant in Detroit; the R & V Wagner Ordnance Co., of East Moline, Ill., as assistant chief inspector and night superintendent; and by the Remy Electric Division of the General Motors Corp., at Anderson, Ind., as production engineer in charge of inspection and superintendent of the machine division.

In 1920 Mr. Smith joined in the formation of the Anderson Implement Development Co., of Anderson, Ind., to carry on experimental work on a tractor of his own design to which he devoted a number of years.

Mr. Smith subsequently became chief inspector and later supervisor of material inspection for the Lexington Motors Co., of Connersville, Ind. In 1924 he joined the Columbia Motors Corp., of Detroit, as superintendent, a post which he held four years. His last connection, made in 1926, was with the Watson Stabilator Co., which later became the John Warren Watson Co., with which he devoted his energies to sales work, in which he became very successful.

Louis Schwab

THE passing of Louis Schwab, president of Stevens Walden Worcester, Inc., of Worcester Mass., to his final rest is announced with sincere regret.

Mr. Schwab was born in Newark, N. J., 53 years ago, and received a general education in the schools of his native city. His early business training fitted him for an executive position, and at the age of 28 he became executive manager and controlling partner of Stevens & Co., of New York City, manufacturers of wrenches, tools, and maintenance equipment. He continued as president of the Stevens Co. until 1927, when a combination of the Walden-Worcester Co., of Worcester, Mass., and the Stevens Co. resulted in a change of the company name to the Stevens Walden Worcester, Inc. Mr. Schwab was made vice-president and director of sales for this company, and continued active in its affairs until his death, on Oct. 27.

Mr. Schwab became an Associate Member of the Society in 1918 and was for a period of years a member of the Metropolitan Section and later of the New England Section until the time of his death.

René Cozette

ANNOUNCEMENT has been made with deep regret that René Cozette, inventor of the Cozette carbureter and supercharger, met accidental death at the autodrome in Monthlery, France, on Aug. 20.

M. Cozette, who, at the time of his death was only 34 years of age, was born in Amiens, France, and received the major part of his education at the Lycee de Tours, where he was a student until 1912. For two years he studied electrical engineering, but joined the French army immediately at the outbreak of the war, and in 1917 became a member of the technical section of the French Flying Corps, with the rank of Sergeant. In this section he made studies and tests of airplane engines.

The following year he entered the services of the Lizaire Berwick Motor Car Co. as an engineer, but in 1919 became administrator and technical director of his own company, the Cozette Carbureter & Supercharger Co., in Neuilly, France. In that year he also became a member of the Society of Civil Engineers of France.

M. Cozette was elected a Foreign Member of the Society of Automotive Engineers in November, 1926.

Personal Notes of the Members

Commander Wilson Leaves Navy

After 20 years of engineering activity in the United States Navy, Commander Eugene E. Wilson has resigned from that branch of the Government Service to become the vice-president and general manager of the Hamilton Standard Propeller Corp., of Pittsburgh, the home office of which will be his headquarters. This corporation, which is a division of the United Aircraft & Transport Corp., resulted from consolidation of the Hamilton Aero Mfg. Co., of Milwaukee, and the Standard Steel Propeller Corp., of Pittsburgh.

Commander Wilson graduated from the United States Naval Academy in 1908. He amplified his engineering studies in 1915 by a course in engineering at Columbia University. His experience in aviation began in 1919, when he became officer-in-charge of the Aviation Mechanics' School at Great Lakes, Ill., where a large percentage of the aviation personnel of the Navy was trained.

Commander Wilson later served as engineering officer of the Aircraft Squadrons Scouting Fleet and in 1924 was ordered to the Bureau of Aeronautics of the Navy Department, in charge of aircraft engine design. Under his supervision the Navy's part in the development of the modern air-cooled aircraft engines was carried on. Commander Wilson directed the Navy Department's share in the perfection of the Wright Whirlwind engines and the Pratt & Whitney Hornet and Wasp series. He was also active in the advances made in the high-speed water-cooled engine field. In addition, he sponsored improvements in aircraft engine accessories.

In the latter part of 1926, Commander Wilson took charge of the designing section of the Bureau of Aeronautics and supervised the design and production of the fighting, scouting, and torpedo and bombing planes now based on the aircraft carriers U. S. S. Langley, Saratoga and Lexington. In the autumn of 1927, he became Chief-in-Staff to Rear Admiral Joseph M. Reeves, the Commander of the Aircraft Squadrons of the Battle Fleet, and in this capacity assisted Admiral Reeves in the organization, training and operations of that command. The organization comprised 15 squadrons of airplanes, totaling more than 200 carrier-based aircraft.

When Admiral Reeves was transferred to the General Board in the City of Washington, Commander Wilson became the Battle Fleet Aviation Officer and aide to Admiral L. M. Nulton, Commander-in-Chief of the Battle Fleet, where he was serving at the time of his resignation.

EUGENE E. WILSON

Commander Wilson was elected a Service Member of the Society in April, 1926. He presented a paper at the Aeronautic Meeting in 1926, entitled, Aircraft Engines in Naval Aircraft, which was published in THE JOURNAL for September, 1926, p. 221. Another paper, delivered at the Aeronautic Meeting in 1928, entitled, Naval Air-Tactics and Aircraft Design, was printed in THE JOURNAL for October, 1928, p. 353. At the November, 1929, Aeronautic Meeting, Commander Wilson presented a paper under the title, The Trend in Aircraft-Engine Design, which is to be published in THE JOURNAL in the near future. Commander Wilson is chairman of the Aeronautic Division of the Southern California Section.

Robert Insley Now Vice-President

Announcement is made that Robert Insley, formerly chief engineer of the aeronautical division of the Continental Motors Corp., has been elected vice-president of the Continental Aircraft Engine Co., of Detroit.

Mr. Insley has been a member of the Society since 1921, when he entered as a Junior member. In October, 1924, while serving with the Air Service Engineering Division, at McCook Field, Dayton, Ohio, he was transferred to the grade of Service Member. His complete work and Society activities were reviewed in a Personal Note published in the October, 1928, issue of THE JOURNAL, p. 38 of the advertising section. During the year 1929, Mr. Insley served on the Aircraft Engine Division of the Standards Committee.

A most interesting paper entitled, Demands of Air-Passengers Challenge Engine-Builders' Ability, was presented at a meeting of the Washington Section in November, 1929, and is published in this number of THE JOURNAL, p. 38.

Emmert Made President of General Motors Radio Corp.

Roger J. Emmert, chosen as president and general manager of the new General Motors Radio Corp., assumes his new position reinforced with a wealth of experience in precision manufacturing, particularly in the electrical field. Now 35 years of age, he has been connected with the General Motors organization for 10 years and was president of the Delco Products Corp., of Dayton, Ohio, a General Motors subsidiary, when selected to direct the activities of the corporation in the radio industry when it purchased the business of the Day-Fan Electric Co., of Dayton.

Mr. Emmert's ability as an engineer and expert in production methods is well known in automotive circles. As president and general manager of the Delco Products Corp. and before that as a member of the manufacturing staff of the Delco-Remy Corp., he had much to do with establishing the production systems now in use in those plants.

Born in Piqua, Ohio, Mr. Emmert attended the public schools of that city and then entered the Case School of Applied Science at Cleveland, specializing in electrical engineering, and was graduated in 1916 with the degree of Bachelor of Science.

He began his commercial career with the Westinghouse Electric & Mfg. Co. in the capacity of graduate student engineer. In 1919 he entered the service of the General Motors Corp., his first position being with the Remy Electric Co., at Anderson, Ind.

When the Remy Electric Co. was combined with the Dayton Engineering Laboratories in 1926, under the corporate name of Delco-Remy Corp., Mr. Emmert's position was that of factory manager; and when this company was divided in 1928 and the Dayton plant incorporated as the Delco Products

Corp., he was made president and general manager.

Mr. Emmert is a Member of the Society and has served as a member of the Nominating Committee. He is also a member of the Dayton Section, S.A.E., of which he was Vice-Chairman in 1928, and of the Dayton Engineers' Club.

Smith Heads White Production

George W. Smith, Jr., has been advanced from the post of technical assistant to the vice-president and general manager of the White Motor Co., of Cleveland, to the position of vice-president in charge of production.

A member of the Society since 1917, Mr. Smith has been active on committees and served during 1928 as a member of the Passenger Car Division of the Standards Committee, being then connected with the Victor Talking Machine Co.

He first entered the employment of the White organization in 1913 as research engineer and returned to it after several years with the Victor Company.

Mr. Smith has presented two papers before the Society. His address on Motor Trucks in War and Commercial Service appeared on p. 218 of THE JOURNAL for March, 1918, and his paper on Evolution of the Present-Day Motorcoach Chassis delivered before the Metropolitan Section, was printed in the June, 1928, issue of THE JOURNAL, p. 395.

Attendu Opens Consulting Office

André C. Attendu, who was consulting engineer of the Attendu Engines Co., Ltd., of Montreal, has established offices at 266 St. James Street, West, Montreal, Canada, as a consulting engineer on high-speed heavy-oil engines. The affairs of the Attendu Engine Co., which was only a holding company, were wound up last May and Mr. Attendu bought back all of his patents.

Mr. Attendu has had 25 years of practical experience in the design and construction of heavy-oil engines in Europe and especially on light-weight oil engines on this continent. He was responsible for the design and construction of a two-cylinder experimental aviation oil-engine for the United States Navy in 1924, and presented a paper entitled, The Attendu Heavy-Oil Engine, at the Annual Meeting of the Society in 1926, published in THE JOURNAL for February, 1926, p. 214, in which these engines were illustrated and described. He is the owner of various patents issued in the United States and in other countries relating to fuel injection, compression control, starting and flexibility of high-speed heavy-oil engines.

Mr. Attendu has been a Member of the Society since 1924, and is also a member of the Buffalo Section. Born at St. Etienne, France, in 1886, and being graduated from the Mining School there in 1909 with the degree of Mechanical Engineer, he studied Diesel-engine development as a student under Dr. Diesel's supervision from 1906 to 1908 and during the following year gained practical experience in the designing and testing of aviation engines, automobiles and especially rear axles at the LeRhone and LaBuire automobile works.

Coming to Canada in 1910, Mr. Attendu went to work in the eastern division of the Canadian Pacific Railway under the chief engineer and at the same time worked on the design of a small flexible Diesel engine of his own. During the war period he served as engineer of the St. Charles Chemical Co. on the manufacture of synthetic picric acid and acetone under his own formulas for the British Government and as production engineer for Lymberner, Ltd., in the making of shells for the United States. From 1920 to 1927 he was president successively of the Automotive Engineering Co. and its successor, the Eastern Engineering Co., of Montreal, engaged in the designing, building and testing of high-compression flexible engines and heavy-oil engines. In 1927 he became consulting engineer of the Attendu Engines Co.

Mock Named Bendix Engineer

Recently F. C. Mock, who has long been connected with the Stromberg Motor Devices Co., of Chicago, joined the South Bend, Ind., engineering department of the Bendix Aviation Corp., of which organization the former Stromberg concern has since become a subsidiary.

Mr. Mock ranks among the engineers who have been active in the automotive field since the early days of the automobile industry. After studying at Western Reserve University, Cleveland, he entered the employment of the F. B. Stearns Co. in September, 1903, as draftsman, and remained in that position until 1905, when he became affiliated with the Royal Tourist Car Co. as draftsman and assistant engineer. He later held positions as engineer with the Consolidated Motor Car Co., the Stoddard-Dayton Co., the Thomas B. Jeffery Co., and the McDuffee Automobile Co., of Chicago.

F. C. MOCK

Mr. Mock's connection with the Stromberg Motor Devices Co. began in 1912, when he entered the organization as consulting engineer, becoming experimental engineer in 1914 and research engineer in 1915.

Since Mr. Mock was elected a Member of the Society in July, 1911, he has served repeatedly on committees and also delivered many papers at meetings. He has been a member of the Research Committee since 1923, of the Publication Committee during 1923 to 1925, of the Fuels Subcommittee since 1927, and of the Cooperative Fuels Research Steering Committee in 1928. During 1924 he served as Chairman of the Mid-West Section, and the following year as Vice-Chairman of the Chicago Section.

Following are the papers given by Mr. Mock which have appeared in publications of the Society; Exhaust Discharge and Exhaust-Manifold Design, p. 256 of the S.A.E. BULLETIN, December, 1913, and p. 69 of TRANSACTIONS, part 1, 1914; A few Points on Carburetion, p. 565 of the S.A.E. BULLETIN, February, 1917; The Fuel Question from the Driver's Viewpoint, p. 249 of the S.A.E. BULLETIN, June, 1917, and p. 347 of TRANSACTIONS, part 1, 1917; Design of Intake Manifolds for Present Heavy Fuels, p. 189 of THE JOURNAL, March, 1920, and p. 983 of TRANSACTIONS, part 1, 1920; Utilization of Present Automotive Fuel, p. 26 of THE JOURNAL, January, 1921; Fundamental Laws of Carburetor Action, p. 85 of THE JOURNAL, August, 1921, and p. 163 of TRANSACTIONS, part 2, 1922; Abstract of Talk at Chicago Fuel Session, p. 3 of THE JOURNAL, January, 1922; The Hot-Spot Method of Heavy Fuel Preparation, p. 27 of THE JOURNAL, July, 1922, and p. 58 of TRANSACTIONS, part 2, 1922 (co-author with M. E. Chandler); The Hot-Spot Method of Heavy Fuel Preparation, supplementary remarks to preceding paper, p. 474 of THE JOURNAL, December, 1922, and p. 58 of TRANSACTIONS, part 2, 1922; Elementary Dynamics of Vehicle Spring-Suspension, p. 37 of THE JOURNAL, July, 1925; Relation of Spring-Suspension to Riding-Qualities, p. 288 of THE JOURNAL, September, 1926; Dual Carburetion and Manifold Design, p. 593 of the S.A.E. JOURNAL, June, 1929.

(*Continued on page* 40)

Applicants Qualified

The following applicants have qualified for admission to the Society between Nov. 10 and Dec. 10, 1929. The various grades of membership are indicated by (M) Member; (A) Associate Member; (J) Junior; (Aff.) Affiliate; (S M) Service Member; (F M) Foreign Member.

ALBINAIK, A. S. (J) assistant body engineer, Cadillac Motor Car Co., Michigan and Clark, *Detroit;* (mail) 14216 Freeland Avenue.

ALLISON, JOHN M., JR. (J) assistant aeronautical engineer, Navy Department, United States Naval Air Station, A. and R. Department, *Lakehurst, N. J.*

ASAI, KIKUSABURO (J) 1702 Geddes Avenue, *Ann Arbor, Mich.*

BAXTER, V. A. (J) lubrication engineer, Midwest Oil Co., 2500 Minnehaha Avenue, *Minneapolis.*

BEALS, ERL VINTON (A) experimental engineer; (mail) 18 Haviland Street, Suite 3, *Boston.*

BENSON, BERNARD H. (J) inspector, construction by contract, Sewer Department, District of Columbia, District Building, *City of Washington;* (mail) Apartment 136, 1900 F Street, N. W.

BLOSS, RICHARD R. (M) vice-president, general manager, International Derrick & Equipment Co., 875 Michigan Avenue, *Columbus, Ohio.*

BOBEK, WILLIAM (M) draftsman, Fisher Body Corp., *Detroit;* (mail) 16601 Washburn.

BOEING, WILLIAM E. (M) chairman of the board, Boeing Airplane Co., 200 West Michigan Street, *Seattle, Wash.;* (mail) P. O. Box 556, Georgetown Station.

BOLLACK, LUCIEN ARMAND (F M) manager, Cifada 12, Rue des Sablons, Paris, France; (mail) 43 Rue Copernic, *Paris XVI°, France.*

BREDE, O. M. (M) director of service, General Motors Truck Co., Pontiac, Mich.; (mail) 176 Suffield Road, *Birmingham, Mich.*

BRIDGES, KENNETH L. (A) special representative, automotive aviation division, American Lithographic Co., Inc., 52 East 19th Street, *New York City;* (mail) 26 Broadway.

BROUGHAM, HERBERT W. (A) proprietor, Jordan Sales & Service Co., 1531 Broadway, *Seattle, Wash.*

BROWN, LOUIS L. (A) production engineer, J. F. Dixon Co., 1835 South Hope Street, Los Angeles; (mail) 417 Elvina Drive, *Glendale, Calif.*

BURKHARDT, CONRAD C. C. (M) engineer, development engineering division, Baldwin Locomotive Works, *Philadelphia;* (mail) 5234 Hazel Avenue.

BURNS, WILLIAM J. (J) assistant foreman, Simons Stewart & Foy, 631 West 57th Street, New York City; (mail) 21 Grenfell Avenue, *Kew Gardens, L. I., N. Y.*

CHAPLINE, GEORGE F. (M) service director, Wright Aeronautical Corp., *Paterson, N. J.*

CHRISTEN, JOSEPH C. (M) head of production control, Willys-Overland Co., *Elyria, Ohio.*

CLARK, M. T. (A) manager, Portland branch plant, Laher Auto Spring Co., Inc., 51 North 15th Street, *Portland, Ore.*

DANIEL, NORMAN HOYLES (M) service engineer, General Motors Products of Canada, Ltd., *Oshawa, Ont., Canada.*

DUDLEY, LYNN B. (A) eastern manager, Campbell-Ewald Co., 292 Madison Avenue, *New York City.*

ELLIOTT, H. A. (M) vice-president, operations manager, Pitcairn Aviation, Inc., Brooklyn, N. Y.; (mail) Pitcairn Aviation, Inc., *Richmond, Va.*

FOSTER, A. H. (M) superintendent, motor traffic department, Toronto Transportation Commission, 35 Yonge Street, *Toronto, Ont., Canada.*

FOX, MERRITT L. (M) manager of research, Houde Engineering Corp., 537 East Delavan Avenue, *Buffalo.*

FROBES, C. R. (A) assistant branch manager, International Harvester Co. of America, 660 East Erie Avenue, *Philadelphia.*

GRAHAM, RALPH G. (A) chief inspector, Alamo Engine Co., *Hillsdale, Mich.*

GRAVES, BERT (A) director of purchases, White Motor Co., 842 East 79th Street, *Cleveland.*

GRIFFIN, WILLIAM (A) garage foreman, Crown Mills, *Portland, Ore.;* (mail) 7106 51st Avenue, S. E.

GRIFFITH, M. O. (M) chief engineer, powerplant, Crosley Aircraft Co., *Cincinnati.*

HANKS, STEDMAN (A) president, Stedman Hanks & Co., Inc., *Manchester, Mass.*

HAWKINS, HARRY (J) engineer, research, General Motors Research Corp., General Motors Building, *Detroit.*

HEATH, J. M. (M) chief draftsman, layout, Mercury Aircraft Corp., *Hammondsport, N. Y.*

HEDBERG, DUDLEY ERIC (A) chief engineer, service manager, Hedberg Bros. Proprietary, Ltd., Tasmania, Australia; (mail) 69 Giblin Street, *New Town, Hobart, Tasmania, Australia.*

HERMANN, OTTO (M) president, chief engineer, Century Rotary Motor Corp., *Canastota, N. Y.*

HOEKSTRA, HAROLD D. (J) chief airplane engineer, Crosley Aircraft Co., Sharonville, Ohio; (mail) Sharon Avenue, *Glendale, Ohio.*

HOLLAND, STEPHEN J. (J) draftsman, Bendix Aviation Corp., 401 Bendix Drive, *South Bend, Ind.;* (mail) 2101 Lincoln Way, West.

HOLST, CARLOS BERNARD (A) manager, mechanical service department, Inter-Mountain Electric Co., *Salt Lake City, Utah.*

HOUSTON, HERBERT W. (A) shop superintendent, experimental and research laboratory, Hughes Tool Co., 1001 North Orange Drive, *Los Angeles.*

HUBBARD, FRANK SIMMONS (M) chief engineer, Berliner-Joyce Aircraft Corp., Dundalk, Md.; (mail) 3313 Crossland Avenue, *Baltimore.*

INDEPENDENT OIL & GAS CO. (Aff.) Tulsa, Okla. Representatives: Atchison, J. R., manager of sales promotion; Craig, L. F., assistant to vice-president; Martin, Villard, vice-president; Rogers, W. W., manager, aviation sales; Wadsworth, J. M., manager of manufacturing.

INGALLS, DAVID S. (A) Assistant Secretary of the Navy, Navy Building, *City of Washington.*

INGLIS, J. W. (J) assistant foreman, electric repair department, Begg Bros., Ltd., *Vancouver, B. C., Canada;* (mail) 1546 Comox Street.

KELLY, RAYMOND D. (M) aeronautical instrument engineer, American Paulin System, Inc., 1220 Maple Avenue, *Los Angeles;* (mail) 10528 Lauriston Avenue.

KREBS, CARO A. (A) owner, C. A. Krebs Oil Co., 421 North Alameda Street, *Los Angeles.*

KUHLMAN, GEORGE W. (A) eastern district manager, Heil Co., Milwaukee; (mail) Heil Co., 31st Place and Star Avenue, *Long Island City, N. Y.*

LA MOTHE, R. F. (J) experimental research engineering department, Waukesha Motor Co., *Waukesha, Wis.;* (mail) 421 West Avenue.

LINDEN, DONALD S. (A) sales manager, lubricant department, R. M. Hollingshead Co., Ltd., 503 Metropolitan Building, *Toronto, Ont., Canada.*

LUCAS, JOSEPH A. (A) superintendent, Winfield carbureter department, Haiden Auto Parts, Inc., 147 12th Street, *Oakland, Calif.*

MAHLE, ERNST (F M) managing director, Elektronmetall Corp., *Stuttgart, Cannstatt, Germany;* (mail) Stuttgart-Rappweg 3.

MOTT, E. PARKER (A) Autosales Corp., Long Island City, N. Y.; (mail) 9460 224 Street, *Queens Village, L. I., N. Y.*

MUNGER, MYRON T. (J) engine designer, Kinner Airplane & Motor Corp., 629 West Colorado Street, *Glendale, Calif.;* (mail) 210 East Garfield Avenue.

OTT, FRANK J. (A) division motor-vehicle inspector, in charge of motor-vehicle maintenance, New Jersey Bell Telephone Co., 540 Broad Street, *Newark, N. J.;* (mail) 975 18th Avenue.

PARKER, FRED W., JR. (J) sales engineering department, Timken-Detroit Axle Co., 100 to 400 Clark Avenue, *Detroit.*

PARRISH, RUSSELL JAMES (M) assistant engineer, chief draftsman, Chevrolet Aircraft Corp., 917 North Capitol Avenue, Indianapolis; (mail) 221 North 20th Street, *Richmond, Ind.*

PUDDY, BEVERLEY B. (J) student, mechanical engineering, University of Toronto, Canada; (mail) 100 Dewson Street, *Toronto 4, Ont., Canada.*

ROGERS, CHARLES E. (A) chief inspector, Alemite Corp., 2650 North Crawford Avenue, *Chicago.*

ROGERS, P. B. (M) assistant chief engineer, Great Lakes Aircraft Corp., 16800 St. Clair Avenue, *Cleveland.*

ROSS, THURSTON H. (M) consulting engineer, chairman, Department of Industrial Management, University of Southern California, *Los Angeles;* (mail) 810 South Spring Street.

SEAMAN, MILTON L. (J) student, University of Maryland, College Park, Md.; (mail) 207 Baltimore Avenue, *Takoma Park, Md.*

SPRINK, I. W. (M) engineer, factory manager, Logan Gear Co., *Toledo, Ohio.*

STAPLES, CHARLES W. (S M) associate engineer, automotive lubrication, Bureau of Standards, Connecticut Avenue and Van Ness Street, *City of Washington.*

STEYR-WERKE A. G. (Aff.) Teinfaltstrasse 7, *Vienna 1, Austria.* Representative: Saaf, Hans V., engineer, general director.

STORY, CHASE S. (M) co-partner, designing wood aircraft-propellers, Story-Gawley Co., 1540 North San Fernando Road, *Glendale, Calif.*

STOVER, GEORGE H. (A) superintendent, maintenance and repairs, Ames Transfer Co., 3441 Kingsbridge Avenue, *New York City;* (mail) 224 Naples Terrace.

SWIFT, STANLEY J. (A) salesman, Cleveland Pneumatic Tool Co., *Cleveland.*

VERETENNICOFF, VLADIMIR A. (J) engineer, General Motors Continental S. A., *Antwerp, Belgium;* (mail) 6 Avenue Rue Albert, Edgehem chez Anvers.

VORHAUER, H. G. (A) engineer, in charge of ground school, Pacific Technical Universary, 2119 Kettner Boulevard, *San Diego, Calif.*

WILCOX JOHN (A) Los Angeles branch manager, C. H. Will Motors Corp., 1726 East Seventh Street, *Los Angeles.*

Applicants for Membership

The applications for membership received between Nov. 15 and Dec. 15, 1929, are listed below. The members of the Society are urged to send any pertinent information with regard to those listed which the Council should have for consideration prior to their election. It is requested that such communications from members be sent promptly.

ABRAMS, DWIGHT RICKARD, draftsman, Chrysler Motors Corp., *Highland Park, Mich.*

ABRAMS, JESS, draftsman, American Chain Co., *Bridgeport, Conn.*

ANDERSEN, HAROLD EDWARD, motorcoach engineer, Studebaker Corp., *South Bend, Ind.*

ANDREWS, HARRY M., captain, Quartermaster Corps., U. S. A., *City of Washington.*

AUG, WILLIAMS F., airplane design, Keystone-Loening Aero Corp., *New York City.*

AUSMAN, JOHN G., instructor, automobile mechanics, Milwaukee Vocational School, *Milwaukee.*

BANNERMAN, REGINALD A., automotive supervisor, Division 6, American Telephone & Telegraph Co., *Cleveland.*

BARRY, A. M., president, St. Lawrence Welding & Engineering Works, Ltd., *Montreal, Que., Canada.*

BAUER, WALTER C., mechanical engineer, Standard Oil Development Co., *Bayway, N. J.*

BEADLE, HAROLD IRVING, New England automotive sales engineer, Tidewater Oil Sales Corp., *Boston.*

BOSS, KENNETH A., lubrication engineer, Willhelm Oil Co., *St. Paul.*

BOULT, C. M. G. D., sales manager, truck department, John McGrath, Ltd., *Sydney, N. S. W., Australia.*

BOYNTON, FREDERICK L., graduate student, Purdue University, *West Lafayette, Ind.*

BOYS, HENRY CECIL, assistant superintendent of design, design department, Royal Arsenal, *Woolwich, S. E. 18, England.*

BREGGER, JOSEPH M., draftsman, Pierce-Arrow Motor Car Co., *Buffalo.*

BRINK, W. S., development engineer, Firestone Steel Products Co., *Akron, Ohio.*

BURCH, FRANK M., quality manager, H. H. Franklin Mfg. Co., *Syracuse, N. Y.*

CALLAHAN, VINCENT T., reserve powerplant engineer, Bell Telephone Laboratories, *New York City.*

CANOOSE, JOHN E., lieutenant, U. S. N., engineer of submarine division, U. S. S. Bushnell, c/o Postmaster, *New York City.*

CASH, V. T., draftsman, Wichita Falls Motor Co., *Wichita Falls, Texas.*

CELLA, JOSEPH A., first lieutenant, Field Artillery, War Department, *City of Washington.*

CHUDAKOV, EUGENE A., chairman, scientific council, Scientific Automotive Institute of the U. S. S. R., *Moscow, Russia.*

COATES, J. EDWIN, propeller engineer, Hamilton Standard Propeller Corp., *Pittsburgh.*

COON, E. E., Westchester Airport Corp., *Armonk, N. Y.*

COOPER, WALTER FRED, special Ethyl gasoline representative, Union Oil Co. of California, *Seattle, Wash.*

COX, JOHN C., engineer, Wagner Electric Corp., *St. Louis.*

CURTICE, H. H., president and general manager, A. C. Spark Plug Co., *Flint, Mich.*

DALAND, ELLIOT, engineer, Keystone Aircraft Corp., *Bristol, Pa.*

DAVENPORT, NOEL, president, Davenport Laboratory, *Hollywood, Calif.*

DEAN, SHIRLEY FAXON, field service representative, Buick Motor Co. of New York, *New York City.*

DEGUISE, GEORGE L., service manager, H. A. Bissonnette, *Lowell, Mass.*

DELL, BRADFORD A., general foreman, Cadillac Motor Car Co., *Detroit.*

DEL PASO, ALEJANDRO, mechanical engineer, airplane designer, Juan E. Azcarate, *Colonia Federal, Mexico, D. F.*

DORR, LEONARD ANTHONY, engineer, Department of Street Railways, *Detroit.*

EARMAN, E. N., in charge of material-returned department, Chevrolet Motor Co., *Flint, Mich.*

EDMISTON, O. N., manager, secretary and treasurer, Benz Spring Co., *Seattle, Wash.*

ERVIN, ROBERT GILPIN, vice-president and general manager, Curtiss-Wright Flying Service of New England, *Boston.*

ESTELLE, FRANK M., staff engineer, Sinclair Refining Co., *New York City.*

FIRTH, DAVID, assistant chief engineer, Marvel Carbureter Co., *Flint, Mich.*

FLETCHER, ROBERT H., experimental engineer, Lycoming Mfg. Co., *Williamsport, Pa.*

GAGE, WILLIAM M., assistant sales manager, Buick Motor Co., *Detroit.*

GRAVES, BENJAMIN P., chief engineer, Brown & Sharpe Mfg. Co., *Providence, R. I.*

GREENEBAUM, LEON C., vice-president, Metropolitan Distributors, Inc., *New York City.*

GROH, CHRISTIAN, chief engineer, Aeronautical Products Corp., *Naugatuck, Conn.*

GUIDER, C. H., automotive engineer, Department of Street Railways, *Detroit.*

HALL, CHARLES LORING, sales engineer, United-Carr Fastener Corp., *Detroit.*

HALL, KENNETH D., engineer, Swallow Airplane Co., *Wichita, Kan.*

HALPIN, THOMAS E., vice-president and general manager, Metal Aircraft Corp., *Cincinnati.*

HEGGEN, EDGAR ALEXANDER, garage superintendent, Carnation Farm Products Co., Inc., *Seattle, Wash.*

HEXEL, LEOPOLD J., chassis layout draftsman, Chrysler Corp., *Detroit.*

HICKS, HAROLD A., aircraft engineer, Ford Motor Co., *Dearborn, Mich.*

HOBBS, W. T., president, Hobbs Mfg. Co., *Fort Worth, Texas.*

HUMPHREYS, W. EASON, manager for Canada, Leyland Motors, Ltd., *Toronto, Ont., Canada.*

HUTCHINS, GEORGE A., superintendent, engine division, White Motor Co., *Cleveland.*

JOHNSON, CARL V., engineer, aeronautical, Bendix Brake Co., *South Bend, Ind.*

JOHNSON, HORACE L., engineering, F. N. Ross, Consulting Engineer, *Detroit.*

KLEMGARD, E. N., assistant chemist, Shell Oil Co., *Martinez, Calif.*

LARRINOA, FRANCIS, field representative, The White Co., *New York City.*

LAWSON, CLARENCE J., president, Casper Oil Corp., *New York City.*

LEARMAN, FRANK A., engineer, Consolidated Aircraft Corp., *Buffalo.*

LEEPER, ALBYN E., mechanic, Studebaker Sales Co., *Chicago.*

LISLE, CLAIBORNE, engine-testing engineer, Hupp Motor Car Corp., *Detroit.*

MAGDEN, JOHN LESLIE, aeronautical engineer, Fokker Aircraft Corp., *Glendale, W. Va.*

MCALLISTER, CHARLES W., sales engineer, automotive engineer, Sinclair Refining Co., *Kansas City, Mo.*

MCCUTCHEON, WALTER WILLIAM, materials engineer, Stearman Aircraft Co., *Wichita, Kan.*

MCFARLANE, COLIN J., representative, Ethyl Gasoline Corp., *New York City.*

MCVITTY, EDWARD WINTON, engineering department, Pan-American Airways, Inc., *New York City.*

MIRANDA, I. J., president, Relay Motors Export Corp., *New York City.*

MURRAY, GEORGE, chief engineer, Tilling-Stevens Motors, Ltd., Victoria Works, *Maidstone, England.*

NELSON, CHARLES E., service superintendent, Harrington Motor Co., *Minneapolis.*

O'BRIEN, JOHN J., president and general manager, Motor & Plane Accessories, Inc., *Detroit.*

PATTERSON, ALLEN B., automotive engineer, The Texas Co., *Chicago.*

PETROFF, A. N., director, School of Aeronautics, University of Wichita, *Wichita, Kan.*

POHL, RUSSELL A., assistant engineer, Kenworth Motor Truck Corp., *Seattle, Wash.*

PRAGER, ALAN O., business manager, Plant 3, Edward G. Budd Mfg. Co., *Philadelphia.*

REARDON, MARION D., branch automobile superintendent, Standard Oil Co. of N. J., *Richmond, Va.*

RENTZ, E. W., JR., service engineer, Sky Specialties Corp., *Detroit.*

RICE, THOMAS M., assistant master mechanic, Calumet Coal Co., *Chicago.*

RUEHL, ERWIN F., assistant chief engineer, oil engineering department, I. P. Morris & De La Vergne, Inc., *Philadelphia.*

SCHMIDT, C. E., engineer, Wagner Electric Corp., *St. Louis.*

SHEPARD, FAYETTE G., lubricating engineer, Pure Oil Co., *Chicago.*

SIRRINE, EARL D., supervisor of motor equipment, Bordens Farm Products Co., *New York City.*

SMITH, WILLIAM OLIVER, draftsman, engineering department, Kinner Airplane & Motor Corp., *Glendale, Calif.*

STANCLIFFE, WILLIAM L., vice-president in charge of sales, A. C. F. Motors Co., *New York City.*

STEPHEN, GEORGE MATHIESON, teacher of automobile mechanics, Board of Education, *Toronto, Ont., Canada.*

STEVENS, NORMAN B., manufacturers sales representative, Firestone Tire & Rubber Co., *Akron, Ohio.*

TAKASHIMA, TAMEO, sales representative, General Motors Japan, Ltd., Tsuru-machi, Minato-ku, *Osaka, Japan.*

TURNER, GUY ALLEN, proprietor, Motor Electric Service Co., *St. Louis.*

WALLACE, J. L., president and general manager, Indianapolis Castings Co., *Indianapolis.*

WEIR, RUDOLPH E., manager, Newton A. Barnett, *Elizabeth, N. J.*

WHELAN, CHARLES M., local sales manager, Aluminum Co. of America, *Detroit.*

WHITE, W. KING, president and treasurer, Cleveland Tractor Co., *Cleveland.*

WILKINSON, LESTER R., experimental engineer and district manager, Jay Mfg. Co., *Chicago.*

Notes and Reviews

AIRCRAFT

Wind-Tunnel Tests on a Series of Wing Models through a Large Angle of Attack Range. Part I—Force Tests. By Montgomery Knight and Carl J. Wenzinger. Report No. 317. Published by the National Advisory Committee for Aeronautics, City of Washington, 1929; 49 pp., illustrated. [A-1]

This investigation covers force tests through a large range of angle of attack on a series of monoplane and biplane wing-models. The tests were conducted in the atmospheric wind-tunnel of the National Advisory Committee for Aeronautics. The models were arranged in such a manner as to make possible a determination of the effects of variations in tip shape, aspect ratio, flap setting, stagger, gap, decalage, sweepback, and airfoil profile. The arrangements represented most of the types of wing system in use on modern airplanes.

The effect of each variable is illustrated by groups of curves. In addition, approximate autorotational characteristics are included in the form of calculated ranges of "rotary instability."

A correction for blocking in this tunnel, which applies to monoplanes at large angles of attack, has been developed and is given in an appendix.

Flight Tests on U. S. S. Los Angeles. Part I—Full-Scale Pressure-Distribution Investigation. By S. J. DeFrance. Report No. 324. Published by the National Advisory Committee for Aeronautics, City of Washington, 1929; 33 pp., illustrated. [A-1]

The investigation reported was conducted at the request of and in conjunction with the Bureau of Aeronautics, of the Navy Department. The purpose was primarily to obtain simultaneous data on the loads and stresses experienced in flight by the U. S. S. Los Angeles which could be used in the design of rigid-airship structures. A secondary object was to determine the turning and drag characteristics of the airship. The stress investigation was conducted by the Navy Department.

The aerodynamic loading was obtained by measuring the pressure at 95 locations on the tail surfaces, 54 on the hull, and 5 on the passenger car. These measurements were made during a series of maneuvers consisting of turns and reversals in smooth air and during a cruise in rough air which was just short of squall proportions.

The results of the pressure measurements on the hull indicate that the forces on the forebody of an airship are relatively small. The tail-surface measurements show conclusively that the forces caused by gusts are much greater than those caused by horizontal maneuvers. In this investigation the tail-surface loadings caused by gusts closely approached the designed loads of the tail structure.

These items, which are prepared by the Research Department, give brief descriptions of technical books and articles on automotive subjects. As a general rule, no attempt is made to give an exhaustive review, the purpose being to indicate what of special interest to the automotive industry has been published.

The letters and numbers in brackets following the titles classify the articles into the following divisions and subdivisions: *Divisions*—A, Aircraft; B, Body; C, Chassis Parts; D, Education; E, Engines; F, Highways; G, Material; H, Miscellaneous; I, Motorboat; J, Motorcoach; K, Motor-Truck; L, Passenger Car; M, Tractor. *Subdivisions*—1, Design and Research; 2, Maintenance and Service; 3, Miscellaneous; 4, Operation; 5, Production; 6, Sales.

Flight Tests on U. S. S. Los Angeles. Part II—Stress and Strength Determination. By C. P. Burgess. Report No. 325. Published by the National Advisory Committee for Aeronautics, City of Washington, 1929; 16 pp., illustrated. [A-1]

The tests described in this report furnished data on the actual aerodynamic forces and the resulting stresses and bending moments in the hull of the U. S. S. Los Angeles during as severe still-air maneuvers as the airship would normally be subjected to, and in straight flight during as rough air as is likely to occur in service, short of squall or storm conditions. The maximum stresses were found to be within the limits provided for in accepted practice in airship design. Normal flight in rough air was shown to produce forces and stresses about twice as great as those produced by the most severe still-air maneuvers. No light was thrown upon the forces that might occur in extreme or exceptional conditions, such as the storm which destroyed the Shenandoah.

The transverse aerodynamic forces on the hull proper were found to be small and irregular. Owing to the necessity of conserving helium, it was impossible to fly the airship in a condition of large excess of buoyancy or weight for the purpose of determining the air-pressure distribution at a fixed angle of pitch. However, every reason exists to believe that in that condition the forces on the actual airship are as close to the wind-tunnel results as can be determined by the present type of pressure-measuring apparatus.

It is considered that the most important data obtained are the coefficients of tail-surface forces and hull-bending moments. These are tabulated in this report.

Tests of Five Metal Model Propellers with Various Pitch Distributions in a Free Wind-Stream and in Combination with a Model VE-7 Fuselage. By E. P. Lesley and Elliott G. Reid. Report No. 326. Published by the National Advisory Committee for Aeronautics, City of Washington, 1929; 18 pp., illustrated. [A-1]

Five adjustable-blade metal model propellers were tested both in a free wind-stream and in combination with a model fuselage with stub wings, at Stanford University under research authorization of the National Advisory Committee for Aeronautics. The propellers are of the same form and cross-section but have variations in radial distributions of pitch. By making a survey of the radial distribution of air velocity through the propeller plane of the model fuselage, it is found that this velocity varies from zero at the hub center to approximately free-stream velocity at the blade tip.

The tests show that the efficiency of a propeller when operating in the presence of the airplane is, over the working range, generally less than when operating in a free wind-stream, but that a propeller with a radial distribution of pitch of the same nature as the radial distribution of air velocity through the propeller plane suffers the smallest loss in efficiency.

Some Studies on the Aerodynamic Effect of the Gap Between Airplane Wings and Fuselages. By Shatswell Ober. Technical Note No. 327. Published by the National Advisory Committee for Aeronautics, City of Washington; 1929, 7 pp., 11 figures. [A-1]

The general result indicated by this study is that, if desirable from any viewpoint, the gap between wing and fuselage can be closed without detrimental aerodynamic effects; and that with a given monoplane the drag is less if the wing is directly on top of the fuselage than if it is parasol.

Cantilever Wings for Modern Aircraft. Some Aspects of Cantilever Wing

(*Continued on next left-hand page*)

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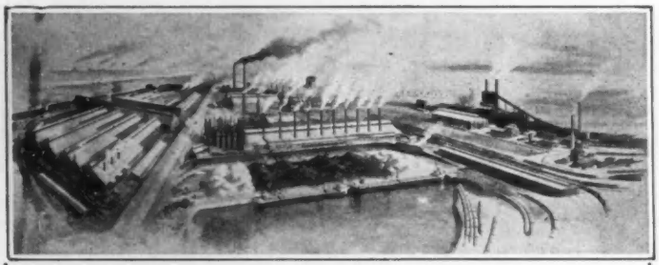

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Notes and Reviews

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Construction with Special Reference to Weight and Torsional Stiffness. By H. J. Stieger. Reprinted from *Aircraft Engineering*, August, 1929. Technical Memorandum No. 538; 14 pp., 8 figures. [A-1]

Unsymmetrical Forces in an Airplane Cell. By R. Vogt. Translated from *Zeitschrift für Flugtechnik und Motorluftschiffahrt*, June 14, 1929. Technical Memorandum No. 539; 9 pp., 4 figures. [A-1]

Technical Details in the Structural Developments of Rohrbach Seaplanes. By Gotthold Mathias and Adolf Holzapfel. Translated from *Zeitschrift für Flugtechnik und Motorluftschiffahrt*, July 15, 1929. Technical Memorandum No. 541; 12 pp., 7 figures. [A-1]

Contribution to the Aileron Theory. By A. Betz and E. Petersohn. Translated from *Zeitschrift für angewandte Mathematik und Mechanik*, vol. VIII, 1928. Technical Memorandum No. 542; 9 pp., 2 figures. [A-1]

The foregoing four Technical Memoranda were issued during November and December by the National Advisory Committee for Aeronautics, City of Washington.

Stabilitätseigenschaften und Steuerbarkeit der Deutschen Landflugzeuge. By Walter Hubner. Published in *Zeitschrift für Flugtechnik und Motorluftschiffahrt*, Oct. 28, 1929, p. 521. [A-1]

While the airplane has progressed to a great extent so far as its efficiency, a concrete and measurable quality, is concerned, its flight characteristics, hard to forecast by mathematical analysis or to judge accurately, have not been developed to the same degree. Acknowledging this condition, the author takes stock of the position of German airplanes in the important qualities of stability and maneuverability. He treats separately of longitudinal, transverse and directional stability and the features of design and operation influencing these characteristics.

No attempt is made to discuss individual makes or models, the author explains, as to do this would involve too extensive details. Another reason advanced for making this article a broad bird's-eye view is that only slight differences exist between individual German craft.

A good rating is given German airplanes in longitudinal and directional stability. In transverse stability they are said not to have reached so advanced a stage. The author deplores that data necessary for comparison with craft of other countries are lacking. One of the few foreign airplanes to be tested in Germany is the DeHaviland Moth, which is said to possess no better flight characteristics than German models of its same class. To this same plane fitted with Handley Page slotted wings is conceded superior maneuverability in stalled flight, in which respect German airplanes are acknowledged to be backward.

Essai sur la Maniabilité. By A. Volmerange. Published in *L'Aéronautique*, October, 1929, p. 331. [A-1]

What is "maneuverability" in an airplane? After setting forth the many meanings this term has for many people and organizations, the author formulates his own definition: "Maneuverability is the ability to pass easily and rapidly from any one flight state to any other that is foreseeable in the utilization of an airplane."

In his discussion the author excludes all characteristics of interest only in stunt or military flying, concentrating on those pertaining to the security of ordinary aerial navigation. He distinguishes four degrees of maneuverability: the possibility of passing from any one normal flight state to another; the ease of such an operation, in which is included the reliability of the controls; the rapidity of such maneuvers; and the possibility of passing from an abnormal flight condition encountered accidentally to a normal flight state.

(*Continued on next left-hand page*)

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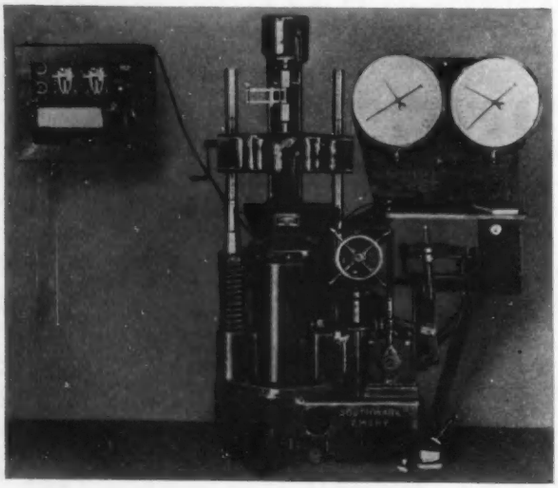

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Notes and Reviews

Continued

While looking forward to the time when maneuverability can be tested with the airplane actually in flight, the author points out the value of wind-tunnel tests and deprecates the criticisms that have been made of them.

Each degree of maneuverability is discussed separately and in detail, with a view to showing that this important characteristic is no longer in the realm of empiricism and may be appraised with the same precision as other qualities of the airplane and to developing the specific criteria that should be included in such an appraisal.

Speed. By T. S. Denham. Published by the Pilot Press, London, 1929; 112 pp., illustrated. [A-3]

In recording the history of the Schneider Trophy, the author has chosen a style intended to attract and enthuse the English public and to urge continued Government assistance to the undertaking. This financial assistance, the author contends, is largely the deciding factor in determining the winner of the trophy but has only of late been recognized by the English Government, despite England's special need for the seaplane in preference to the land plane.

Details and dramatic incidents as well as technical advances in design are covered in this report of the races from 1912 to 1928, the book having been published before the latest British triumph. The value of the publicity and prestige that accompany the winning of records is stressed, and the writer cites, as an example, the selling advantage gained by the British motorcycle manufacturers through the racing successes of British machines.

Americans might be inclined to quarrel with Mr. Denham on one of the points in his summary of the gains to industry through racing; namely, the statement that it discourages mass-production methods, "which, however successful they may be in some industries, are undoubtedly bad for aviation because they hamper progress."

Transport Aviation. By Archibald Black. Published by Simmons-Boardman Publishing Co., New York City, 1929; 348 pp., illustrated. [A-4]

The author of Civil Airports and Airways has turned his attention again to the subject of his earlier book and brought out this completely revised and enlarged edition. We could have no more striking commentary on the rapidly changing status of the aviation business than the author's statement that, "Most of the material in the first edition (although only three years old) had become so obsolete that it was discarded completely and new information substituted."

In collaboration with a group of specialists in various branches of air transportation, Mr. Black has produced a valuable source book with a scope that makes summarizing difficult. A survey of the air-transport business today; trends in the field; the governing factors of future developments; operation requirements, problems, and costs; and equipment, personnel, law and insurance, are only a few of the many phases discussed and analyzed. It is a comprehensive and timely volume.

CHASSIS PARTS

Tire Design with Relation to Comfort, Flexibility, and Wear. By T. M. Knowland. Published in *Rubber Chemistry and Technology*, October, 1929, p. 479. [C-1]

The paper summarizes the more important factors in tire design that exert a controlling influence on tire comfort, flexibility and wear, with a view to giving a clearer insight into some of the simple mechanics and economics of tire behavior.

The use of the pneumatic tire as a shock-absorber is discussed, and a comparison of the relative rates of de-

(*Continued on next left-hand page*)

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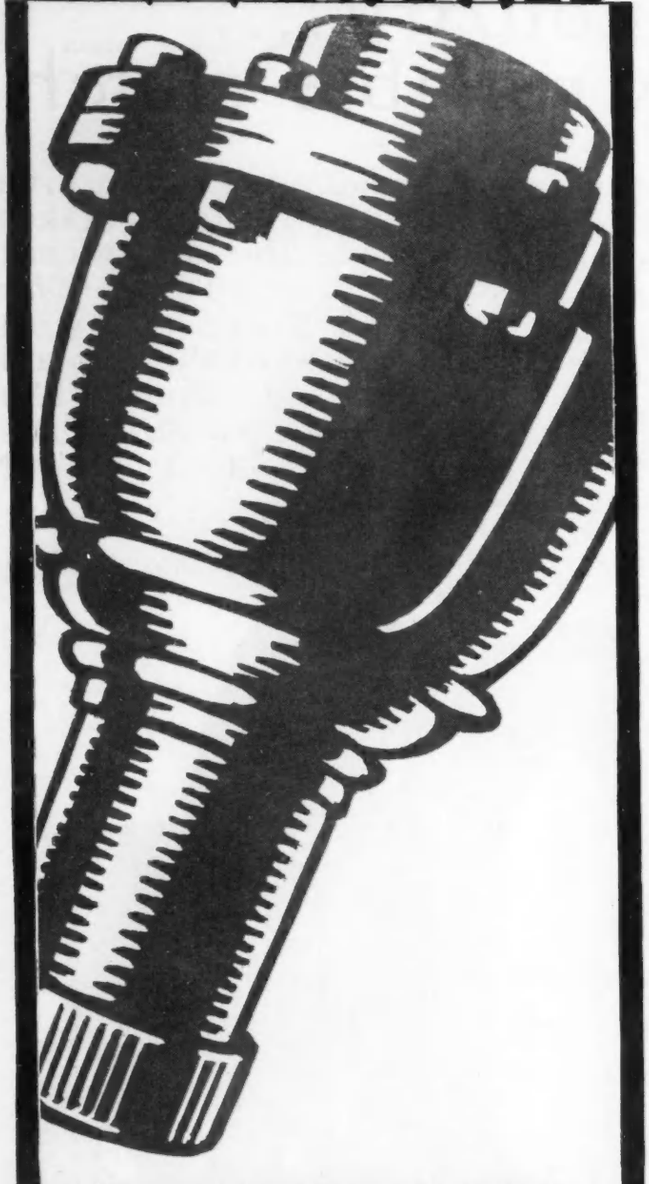
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Notes and Reviews

Continued

flection of a 6-in. tire and a 40-in. spring is shown graphically. Deflection, percentage deflection and rate of deflection are also explained with the aid of diagrams, and the statement is made that tire life is largely determined by the number and severity of the flexures to which the carcass is subjected; and the severity of these flexures is in turn proportional to the percentage of deflection at which the tire is operated.

The author declares that it has been repeatedly demonstrated that for every percentage increase in carcass deflection above 12 per cent there is a rapid and proportionate decrease in tire-carcass life. The money loss amounts to approximately $0.001 per mile, he declares, based on the present retail price schedules and on sizes ranging from 4.40 to 6.75.

The stresses and abrasive wear to which the tire is subjected are analyzed and reduced to formulas.

The Radially Tapered Disc Spring. By W. A. Brecht and A. M. Wahl. Presented at the Annual Meeting of the American Society of Mechanical Engineers, Dec. 2 to 6, 1929, New York City. [C-1]

The advantages of disc springs, both flat and dished, having radially tapered cross-sections are discussed. Equations are developed by approximate and more exact methods for calculating the strength and flexibility of such springs. The theoretical work is checked up by numerous tests, including both strain and deflection measurements. The conclusion is reached that these springs utilize the material about as efficiently as do transverse elliptic springs but not as efficiently as helical springs. On the other hand, these springs have certain advantages over helical springs which make them preferable for certain applications.

Natural Frequency of Gears. By R. E. Peterson. Presented at the Annual Meeting of the American Society of Mechanical Engineers, Dec. 2 to 6, 1929, New York City. [C-1]

The natural frequency of machine parts as related to the noise problem in machine operation is discussed in a general manner.

Vibration phenomena in discs and rings are described in detail, following which the author discusses the manner in which a gear vibrates at its natural frequency.

An empirical formula is given for the natural frequency of a gear in terms of its dimensions and material.

ENGINES

The Effect of Supercharger Capacity on Engine and Airplane Performance. By O. W. Schey and W. D. Gove. Report No. 327. Published by the National Advisory Committee for Aeronautics, City of Washington, 1929; 20 pp., illustrated. [E-1]

Supercharging has already demonstrated its value as a means of improving the performance of an airplane at moderate and high altitudes. To obtain a maximum increase in the performance of an airplane designed to meet definite service requirements, it is necessary that a supercharger of the proper capacity be selected.

The effect of different supercharger capacities on the performance of an airplane and its engine was investigated by the staff of the National Advisory Committee for Aeronautics at Langley Field, Va. The tests were conducted on a DH4-M2 airplane powered with a Liberty-12 engine. In this investigation four supercharger capacities, obtained by driving a Roots-type supercharger at 1.615, 1.957, 2.4, and 3 times engine speed, were used to maintain sea-level pressure at the carbureter to altitudes of 7,000, 11,500, 17,000, and 22,000 ft., respectively.

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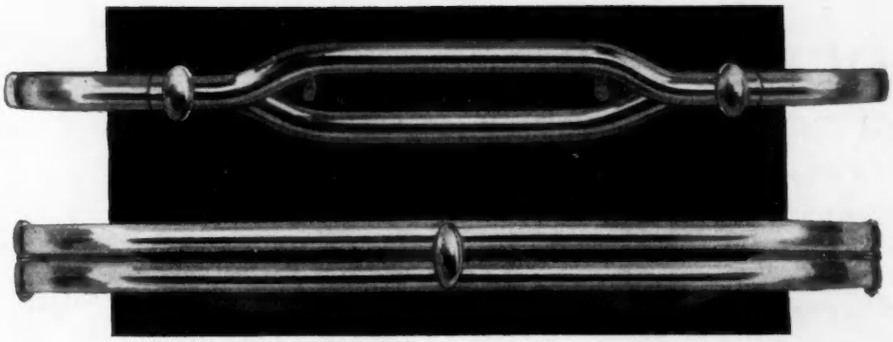

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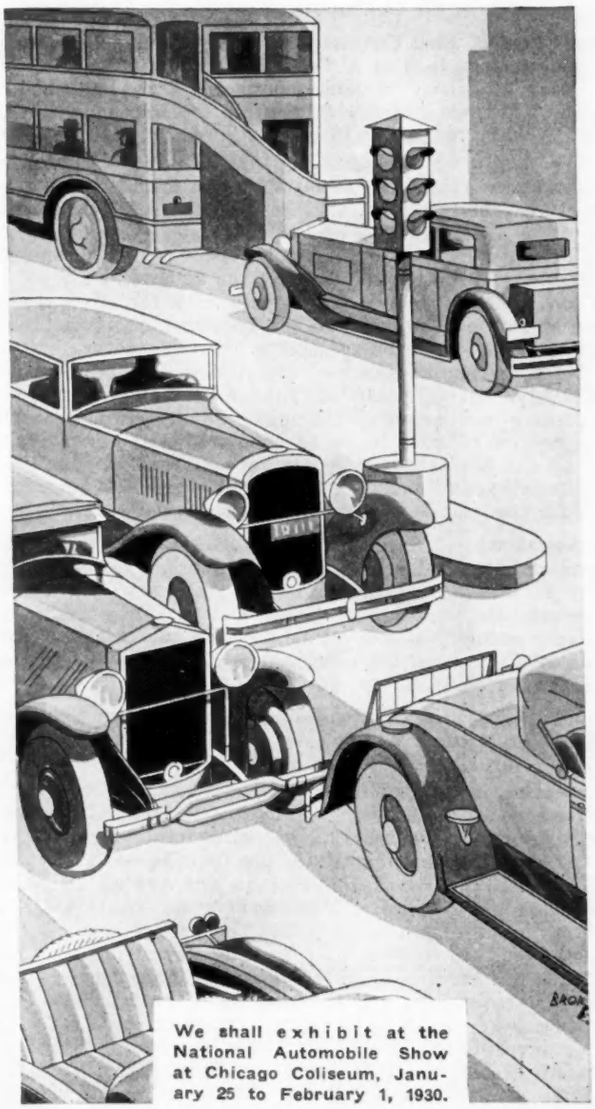

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Notes and Reviews

Continued

The performance of the airplane in climb and in level flight was determined for each of the four supercharger drive-ratios and for the unsupercharged condition. The engine power was measured during these tests by means of a calibrated propeller.

It was found that very little sacrifice in sea-level performance was experienced with the larger supercharger drive-ratios as compared with performance obtained when using the smaller drive-ratios.

The results indicate that further increase in supercharger capacity over that obtained when using the 3:1 drive-ratio would give a slight increase in ceiling and in high altitude performance, but would considerably impair the performance for an appreciable distance below the critical altitude.

As the supercharger capacity was increased, the height at which sea-level high speeds could be equaled or improved became a larger percentage of the maximum height of operation of the airplane.

The Effect of Fuel Consumption on Cylinder Temperatures and Performance of a Cowled Wright J-5 Engine. By Oscar W. Schey. Technical Note No. 328. Published by The National Advisory Committee for Aeronautics, City of Washington, 1929; 18 pp., 9 figures. [E-1]

The results of the tests are presented in the form of tables and curves. The tables show the cylinder-head temperatures obtained with each jet and the location of each of the 69 thermocouples used, the air speed, engine speed, brake horsepower, specific fuel consumption, barometric pressure, and the temperatures of the carbureter air, oil in, oil out, and air-stream. The figures show the effect of the rate of fuel consumption on the temperature of the front and rear spark-plug bosses and on the temperatures of the cylinder barrel, the effect of the reduction in temperature obtained on several points of cylinder No. 1 by enriching the mixture, a comparison of the spark-plug-boss temperatures obtained on cylinder No. 3 as compared with the average for all the unshielded cylinders for different rates of fuel consumption, and the effect of varying the fuel consumption on the engine power and engine speed.

Mathematical and Experimental Investigation of Heat Control and Power Increase in Air-Cooled Aircraft Engines. By F. Gosslau. Translated from *Zeitschrift für Flugtechnik und Motorluftschiffahrt*, Oct. 7, 1928. Technical Memorandum No. 540. Published by the National Advisory Committee for Aeronautics, City of Washington, November, 1929; 12 pp., 25 figures. [E-1]

Emploi des Huiles de Goudron de Houille non Volatiles dans les Moteurs à Explosion. Published in *Le Génie Civil*, Nov. 9, 1929, p. 458. [E-1]

From his investigation into the possibility of using the less volatile of the coal-tar oils as fuel in internal-combustion engines, the author draws unfavorable conclusions. The engine must be adapted to the fuel, he asserts; it is not sufficient to adapt the fuel to the engine by such expedients as vaporizers, atomizers or air heaters.

A device developed to facilitate the use of heavy fuels is described. It consists of a heating device mounted on the inlet manifold and an atomizer. The heating device utilizes exhaust gas, and the amount of heat provided is controllable.

The investigation made with the use of this device, included, besides other features, carefully regulated comparative fuel-consumption tests. The functioning of the engine using the heavy fuel was found to be unsatisfactory, the particular faults being poor acceleration and deceleration, lack of flexibility, dilution of the crankcase-oil and fouling of the spark-plugs and combustion-chambers. A volatile fuel had to be used in starting and warming up the engine, which constituted an additional inconvenience.

(Continued on next left-hand page)

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Notes and Reviews

Continued

The Design of Dynamically Balanced Crankshafts for Two-Stroke-Cycle Engines. By P. Cormac. Published in *Engineering*, Oct. 11, 1929, p. 458. [E-1]

The author points out the importance in the design of multi-cylinder-engine crankshafts of crank sequence and cylinder spacing which will produce an engine having good inertia balance. He states that in general, with two-stroke engines, the best balance is obtained only when a definite firing order is combined with a special spacing of the cylinders. Where, as with engines of less than five cylinders in line, it is not possible to obtain complete primary balance, the out-of-balance effects should be minimized by using suitable counterbalance masses on the crankshaft which will vary with the number of cylinders and crank arrangement.

Formulas for the counterbalance masses for a variety of groupings of cylinders up to 16 are worked out and explained.

Recent Developments in Aircraft Engines. By John H. Geisse. Presented at the Annual Meeting of the American Society of Mechanical Engineers, Dec. 2 to 6, 1929, New York City. [E-1]

Improvements in engine design for airplanes during the last year have been gradual, states the author. Production facilities now exceed the demand and developments should come more rapidly. There has been a trend during the year toward the in-line air-cooled types of engine. However, the radial type still possesses a number of advantages. Manifolding in radial engines is a problem, owing to the impossibility of getting symmetry except in nine-cylinder types, in which thru three-way manifolds can be used.

Cylinder construction with a steel sleeve having machined cooling fins and a cast-aluminum head is accepted rather generally, and a trend is evident toward the use of heat-treated alloys for the crankcase. Many interesting developments in the Diesel types of airplane engine are predicted by Mr. Geisse.

Torsional-Vibration Dampers. By J. P. Den Hartog and I. Ormondroyd. Presented at the Annual Meeting of the American Society of Mechanical Engineers, Dec. 2 to 6, 1929, New York City. [E-1]

In this paper the authors present a theory of the action of friction dampers on systems in torsional resonance, with special application to gas and Diesel-engine installations. Model tests carried out to check this theory are described. The effect of introducing springs in the Lanchester damper is also discussed.

Untersuchungen der Schwingungserscheinungen in der Expansionslinie bei Raschlaufenden Verbrennungs-Kraftmaschinen. By H. Kissinger. Published in *Automobiltechnische Zeitschrift*, Oct. 31, p. 670, and Nov. 10, 1929, p. 701. [E-1]

To what causes may the breaks in the expansion line of the indicator diagram of a detonating engine be justifiably attributed? To answer this question the theoretical and experimental investigation described in this article was conducted.

Three possibilities examined were: the vibrations caused by the explosion of the charge, those incident to the elastic system of the indicator, and, finally, those arising from inertia effects in the cylinder. In the experimental research, a single-cylinder engine and an optical indicator of the OSA type were used.

The data disclosed by the investigation lead to the conclusion that the irregularities of the diagram are directly due to the explosion and vary with the fuel used. The frequency of this type of vibration ranged from 42,700 to 31,000 and from 62,500 to 32,000 per min., while for those

(Continued on next left-hand page)

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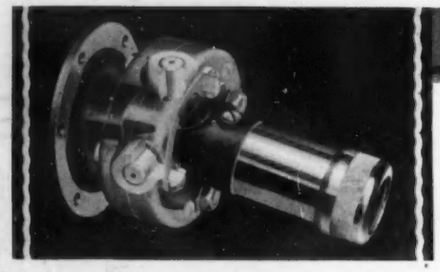

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Notes and Reviews

Continued

arising from the indicator the frequency from 120,000 to 104,000 was too high, and for those due to inertia effects the frequency, from 9700 to 5300, was too low to be considered as entering into the phenomenon under study.

Commercial Applications of High-Speed Oil Engines. By C. H. Gibbons. Published in Oil and Gas Power division, *A. S. M. E. Transactions*, vol. 51, No. 20, May-August, 1929, p. OGP 51. [E-4]

The author discusses the possible application of high-speed Diesel engines to aircraft, trucks, and tractors, and gives details of their application to locomotives, generating sets for city buildings, cotton ginning, oil-well drilling, pipe-line pumping, irrigation, marine propulsion, air compressors, dredges, excavating machinery, and so forth. He also deals with medium-speed applications and gives data for use in estimating power costs.

MATERIAL

Le Problème de la Corrosion des Métaux Ferraux. By V. Sallard. Published in *La Technique Moderne*, Nov. 1, 1929, p. 673. [G-1]

In the struggle against corrosion, the resistance opposed by the metal itself constitutes, from all evidence available, a particularly active defense element. Other means of protection, such as painting and metallic coatings, have valuable qualities in their respective fields of application and will be studied in later articles. The present article is devoted to auto-protection, particularly in irons and steels, the current use of which runs into considerable tonnage.

After expounding the accepted theories explaining the phenomena of metallic corrosion, the author examines a number of rust-resisting ferrous metals, discussing their general characteristics and giving results of corrosion tests. The metals selected include Armco iron; steels containing copper as an alloying element; steels known under the trade names of Apso, Durapso, and Cementapso; high-silicon steels; and chromium steels.

In conclusion, the author praises the progress made so far in the field of auto-protection and calls attention to the research that is still being continued, especially that tending toward the improvement of structural steels by the addition of small amounts of alloying metals.

Service Characteristics of Diesel-Engine Lubricating Oil. By A. E. Flowers and M. A. Dietrich. Presented at the Annual Meeting of the American Society of Mechanical Engineers, Dec. 2 to 6, 1929, New York City. [G-2]

Diesel-engine lubrication has presented difficulties owing to the high compression-pressures, the high temperatures, and the great weight of engine parts. Dr. Diesel himself stated that his first engine ran only three revolutions, when first started, before stoppage was caused by the seizing of the piston-rings. A progressive record of the characteristics of the lubricating oil in use in a Diesel engine in regular continuous service over a period of nearly a year forms the basis for this paper. The engine referred to was in operation on a combination battery-third-rail Diesel-electric locomotive employed in freight-switching service.

La Raffinage du Pétrole Brut; Nécessité de Construire en France de Nouvelles Raffineries. By E. Legé. Published in *Le Génie Civil*, Sept. 7, p. 223; Sept. 14, 1929, p. 241. [G-5]

Urging that the business of petroleum refining be more strongly established in France, the author states that only three refineries are now operating in that country, while the domestic consumption of petroleum products in the near future probably will be equal to the capacity of about ten

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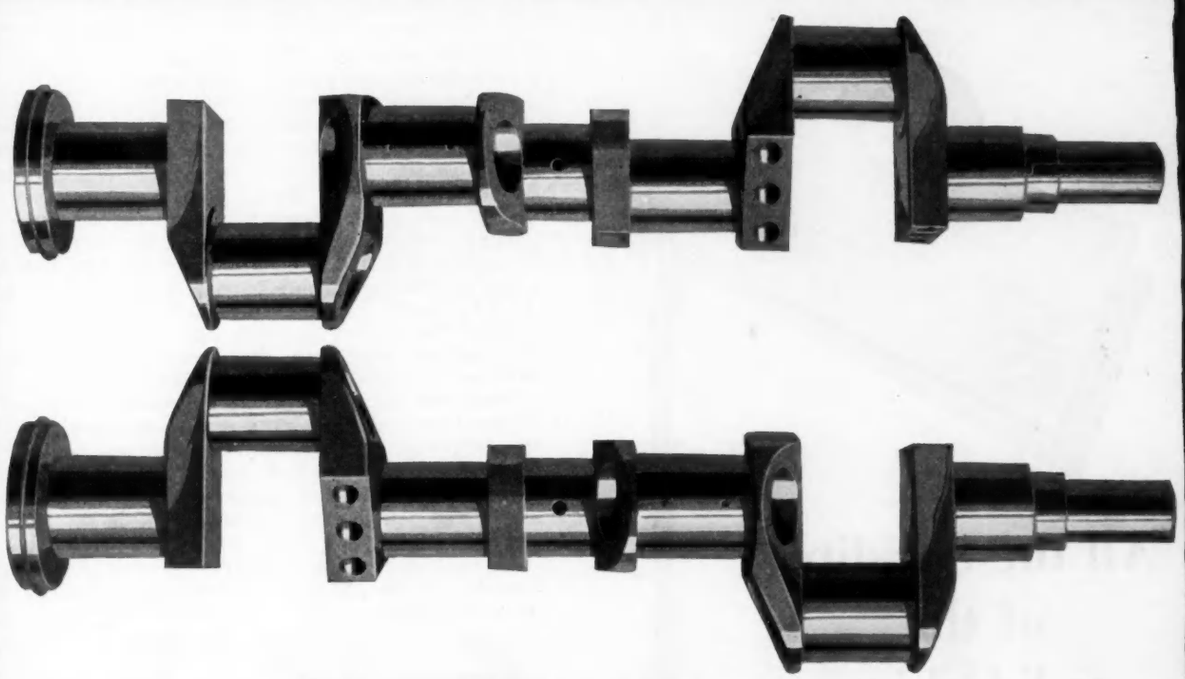

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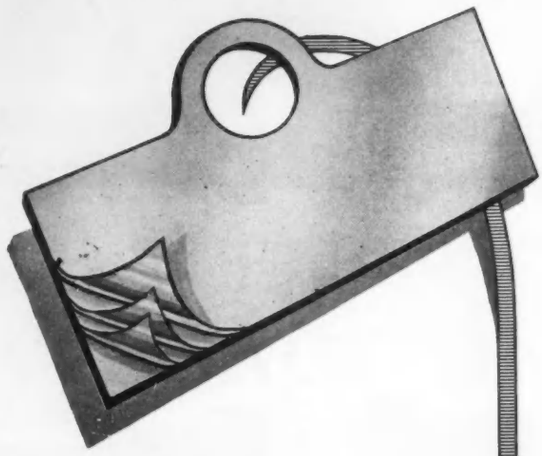

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Notes and Reviews

Continued

fair-sized refineries. The large part of petroleum products now imported into France are in a refined state.

In the matter of raw-petroleum supplies, France is also at a disadvantage. The only domestic wells, those in Alsace, have an annual production of only 70,000 tons, an almost negligible amount compared with the yearly consumption of about 2,215,000 tons. For foreign supplies under her control, France can look only to about one-fourth of the Mesopotamian resources, and these have not been exploited.

To encourage the installation of refineries, France passed a law in 1928 permitting importers to obtain licenses good for 20 years for the importation of raw petroleum, whereas licenses for refined products hold good for only 3 years. Unfavorable conditions that must be faced by French refineries, as contrasted with similar American enterprises, pointed out are: the difficulty, amounting virtually to an impossibility, of obtaining a continuous supply of the same type of crude; the cost of transoceanic transportation of the crude; the relatively high cost of the fuel and chemicals needed in refining; and the present lack of specialized personnel and equipment. One advantage is the cheapness of labor.

An analysis is made of the qualities required in petroleum products, of the principles of refining, and of the types of refining process.

MISCELLANEOUS

Autodictionaer. Published by Dr. Ernst Valentin Verlag, Berlin-Friedenau. 177 pp. [H-1]

This second volume of the trilingual dictionary of automobilism translates the terms from English into French and German. In the first volume, the key list was German, with translations into French and English. For the practical use of those engaged in the automotive industry, 16,000 technical terms and other words used in connection with automobiles and automobile sport are contained in this volume, which was compiled under the auspices of the Association of German Automobile Manufacturers by Benno R. Dierfeld, consulting automobile engineer. This highly specialized and uptodate volume is a convenient and helpful reference book which probably will be appreciated particularly by those who have foreign correspondents, those interested in obtaining patents abroad, and those intending to investigate the automotive situation in France or Germany.

Les Voitures Automotrices Diesel-Electriques du Chemin de Fer d'Appenzell (Suisse). By E. Bard. Published in *Le Génie Civil*, Nov. 9, 1929, p. 460. [H-1]

Opposing to steam power the advantages of greater efficiency and lower personnel requirements, and equalling electricity in flexibility and cleanliness of operation while avoiding the large initial cost of electric installation, the Diesel-electric system for train propulsion has won for itself a trial by the Appenzell railroad in the mountainous canton of that name in Switzerland.

The rail-cars are made by the Société Industrielle Suisse and are powered by Sulzer Diesel engines, while the electrical equipment is furnished by the Ateliers d'Oerlikon. The power cars can haul three cars, accommodating about 150 passengers, up an ascent and control eight on a descent. The engine is of the four-stroke-cycle, mechanical-injection, pre-combustion-chamber type. Each cylinder has its own fuel-pump, controlled by a regulator so that each cylinder receives exactly the requisite quantity of fuel. In addition, an electro-pneumatic control is provided by which the fuel supply can be cut off in descents or at stops, thus making possible important fuel economies.

The engine is directly connected to a continuous-current generator which forms a unit with an auxiliary generator. The hook-up is such that current is conserved and the engine always operates at the same speed.

Further technical details and the results of a run on a test course are given.

(*Continued on next left-hand page*)

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Notes and Reviews

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Men, Money and Motors. By Theodore F. MacManus and Norman Beasley. Published by Harper & Brothers, New York City and London, 1929; 273 pp. and index. [H-3]

Decidedly gossipy and interesting, kaleidoscopic in nature, throwing the light of "pitiless publicity" full in the faces of many of the automotive giants of the last quarter century—Leland, Joy, Chrysler, Fisher, Durant, Willys, White, Franklin, Jewett, Ford, Metzger, Everitt, Brush, Hupp, Olds, Jordan, Winton, Selden, Dodge, Graham, Chapin, Coffin, Chalmers, Couzens, Haynes, Sloan, Kettering, Collins, Briscoe, Bate, Warren, Knudsen, Hawkins, Lee, Lewis, Thomas, Klingensmith, Wilkinson, Bennett, Chandler, Nash, Daniels, Duryea, Anderson, Eames, Drake—Men, Money and Motors, the drama of the automobile, by Theodore F. McManus and Norman Beasley, is well worth reading. Old-timers will enjoy every page of it and incidentally will learn why certain men acted as they did.

An alert observer and a faithful chronicler with a favored seat at the ringside, Mr. MacManus has for many years been in a position to get a good close-up of the struggling champions. He has been in the ring more than once with the hot towel and sponge and a strong bottle of stimulant. Especially well was he acquainted with the Cadillac and Dodge organizations.

Many of the intimate glimpses he gives the reader will bring back in the mind's eye of automotive pioneers a long train of pleasant or perhaps unpleasant thoughts. To the youngsters just entering the industry the book will be inspirational, educating and, best of all, a destroyer of the inferiority complexes which often attack the novice in the early transitional years from college to calling. The book shows the crude beginnings, the conflicts, the doubts and discouragements, the happiness, the successes, the trivialities and triumphs with which the automotive industry is concerned. It is filled with drama, and with the exception of minor points of fact, notably those dealing with Olds, Thomas and Hudson, reflects accurately the great automotive progress of our last quarter century. F. L. F.

Men and Machines. By Stuart Chase. Published by the Macmillan Co., New York City, 1929; 273 pp. and index, illustrated by W. T. Murch. [H-3]

Stuart Chase, the author of The Tragedy of Waste, and, in collaboration with F. J Schlink, of Your Money's Worth, has in Men and Machines added another to a growing list of industrial discussions having to do with the so-called "power age" and modern civilization. Somewhat statistical and not as interesting as Your Money's Worth, possibly because it will not be as clear to the layman and as immediately pertinent as its predecessor, Mr. Chase's book nevertheless contains an excellent survey of so-called prime movers, end-tools, and intermediate transmission devices and their effect on civilization. He has evidently made use of a number of ancient source books, and with his characteristic analytical mind arranged hundreds of interesting facts in an orderly form for more than casual study.

Of interest to the engineer, the book will be useful in arranging one's thoughts as to possible progress and effect. The fact that it contains such a wealth of source material will make its ownership well worthwhile. Correctly distributed in mental pigeon-holes, the volume is a veritable storehouse of mechanical fundamentals. Matters of definition of what is a machine and what is a tool complicate the problem. Fundamental thermodynamic laws, history of machine development, the distribution and classification of the machines of this civilization, questions of consumption and production, labor saving, skill, recreation, robots, all come in for their share of attention. In one of the later chapters, The Balance Sheet, effects manifestly good and effects manifestly bad are carefully tabulated. The reader may draw his conclusion. Emerging, the time-honored question remains, Where do we go from here, boys? F. L. F.

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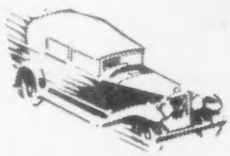

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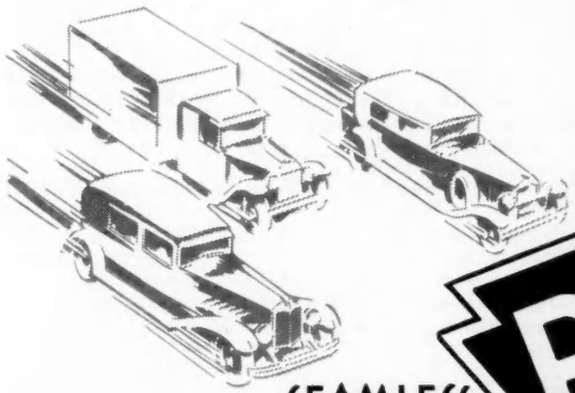

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Notes and Reviews

Continued

PASSENGER CAR

Progress in Design. Published in *The Autocar*, Oct. 25, 1929, p. 807. [L-1]

This summarized review of tendencies in automobile design has been a feature of *The Autocar* show number for several years. The technical staff of that publication outlines the progress during the last 12 months and indicates the probable developments for the near future.

The trend toward an increased number of cylinders is noted. However in England this change is stated to have been confined largely to a gain in the six-cylinder class over the four-cylinder car, since the increase from six to eight cylinders has been made in the main in automobiles of foreign origin, the straight-eight having lost rather than gained adherents among the English manufacturers. This situation is largely due to the horsepower tax, "which has hung like a millstone round the neck of design in this country," asserts the author.

The steady improvement in detail is mentioned, but the need for more work toward securing evenness of gas distribution is emphasized. The performance of aluminum pistons is compared with that of cast-iron pistons, and the conclusion reached that the cast-iron piston, if expected to do the work accomplished by the present-day aluminum piston, would give as much trouble. Under the heading of Ignition, the fact is brought out that, so far as the British market is concerned, coil ignition has established a substantial supremacy over the magneto, more cars now being fitted with coil ignition than with the magneto.

Noticeable detail improvement in four-wheel-brake gear, a movement toward steering mechanism with lower gears, and an increasing use of wire wheels are other features recorded.

A series of graphs showing the rise and fall in popularity of the various features in a car during the last four years is a particularly interesting contribution to the article.

An Engineer at Olympia. By Wilfred G. Aston. Published in *The Autocar*, Oct. 25, 1929, p. 932. [L-1]

This second article in the show number of *The Autocar* covers the technical features of the cars exhibited at Olympia show and is liberally illustrated with photographs and drawings of the outstanding improvements and new devices in evidence there.

The Motor Exhibition at Olympia. Published in *Engineering*, Oct. 18, 1929, p. 490; Oct. 25, 1929, p. 521. [L-1]

For those who desire a more detailed report of the Olympia show, this 12-page review in two issues of *Engineering* is recommended. In addition to summarizing the design trends, the article covers the export question and the outstanding exhibits at the show are described in considerable detail.

Tendances et Progrès de la Construction 1930. By André Caputo. Published in *Omnia*, October, 1929, p. 290, and November, 1929, p. 446. [L-1]

A discussion of the general design tendencies for 1930, a statistical analysis of certain phases of the French automotive industry, and descriptions of novelties both in chassis and accessories constitute the three parts of this article.

The continued popularity of small cars, from 5 to 10 hp., is remarked upon, as well as the hardihood of the eight-cylinder in-line in battering a breach in the wall of the sixes, and the renaissance of the supposedly defunct twelve-cylinder. Greater comfort at medium speeds is the ideal being substituted for continually increasing speeds. The relations between power, weight and speed are discussed. Increasing service facilities being made available and foreign participation in the show are other points commented on.

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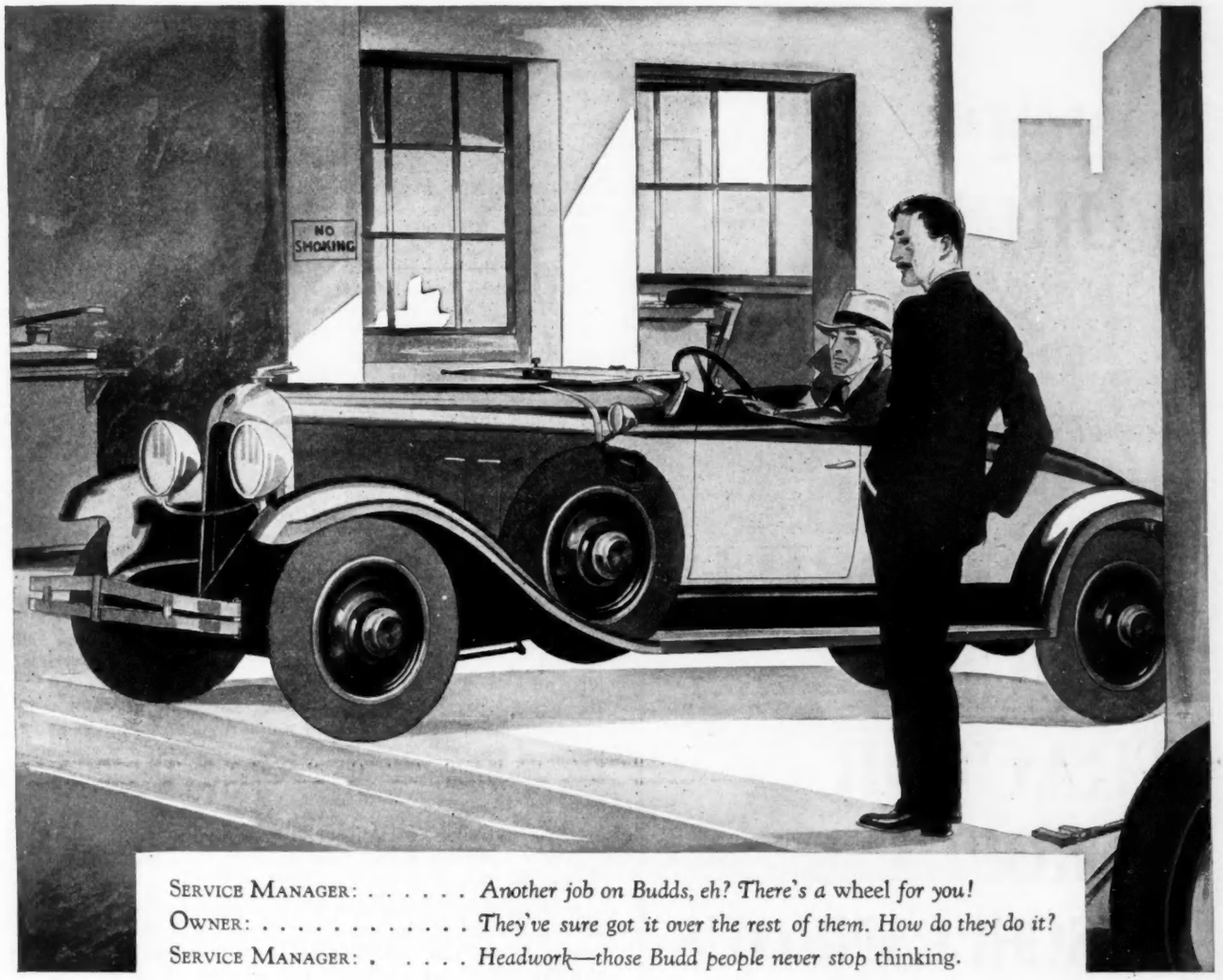


SERVICE MANAGER: . . . . . *Another job on Budds, eh? There's a* wheel *for you!*
OWNER: . . . . . *They've sure got it over the rest of them. How do they do it?*
SERVICE MANAGER: . . . . . *Headwork—those Budd people never stop* thinking.

BUDD DEVELOPED THE BEST DISC WHEEL IN THE WORLD

The Budd-Michelin Disc wheel—the safest automobile wheel in the world

—and they haven't stopped thinking yet!

EVIDENCE

VERY often it takes greater vision to recognize the superior merit of another man's work than to equal his achievement. Budd, for instance, saw the Michelin wheel standing up under the terrific strain of war service. Recognizing on this wheel the theoretically perfect design, Budd entered into alliance with Michelin and made the not inconsiderable contribution of adapting the Michelin design to large-scale production.

BUDD DEVELOPED THE BEST WIRE WHEEL IN THE WORLD

The Budd Wire Wheel—the smartest wire wheel on the roads today

BUDD
WHEEL COMPANY
Philadelphia *Detroit*

DROP FORGINGS

Backed by
47 Years' Experience

AUTOMOBILE
TRUCK
TRACTOR
AEROPLANE
MISCELLANEOUS

Complete Heat Treating and Laboratory Facilities

Capacity 2500 Tons Per Month

Any Type—Any Size—Up to 500 Lbs.

UNION SWITCH & SIGNAL CO.

Drop Forge Division

SWISSVALE, PA.

Notes and Reviews

Concluded

Some of the figures given in the discussion of the French automotive industry are: for 1928, production, excluding 4000 American automobiles assembled in France, 223,600 vehicles; of these roughly 175,000 were passenger-cars, 40,000 small trucks and 9000 large trucks; exports 50,128.

Among the novelties described are the new eight-cylinder Lancia, the Harris-Léon Laisne chassis, the eight-cylinder Röhr with four independent wheels, the eight-cylinder Majestic, controllable radiator shutters, the new Zenith carbureter, headlight controls on the steering-wheel, the Perrier centrifugal oil-purifier, the Radium all-steel piston, and the Bloccyl special light-alloy cylinder-head designed to facilitate the use of high compression-ratios without detonation.

Salon de l'Automobile 1929; Quelque Détails de Construction. By Pierre Maillard. Published in *La Vie Automobile*, Oct. 25, 1929, p. 573. **[L-1]**

Engineers whose technical curiosity is baffled by the usual type of article descriptive of automobile shows, who miss in these generalities, however judiciously and competently written, the minute details that contribute to the perfection of a mechanical part, will welcome this article. The author, figuratively, lifts the hood and climbs under the chassis, and discloses some of the small hidden constructional features of the new models. For instance, some detail of brake-control linkage for 13 cars is described. Line sketches illustrate the discussion.

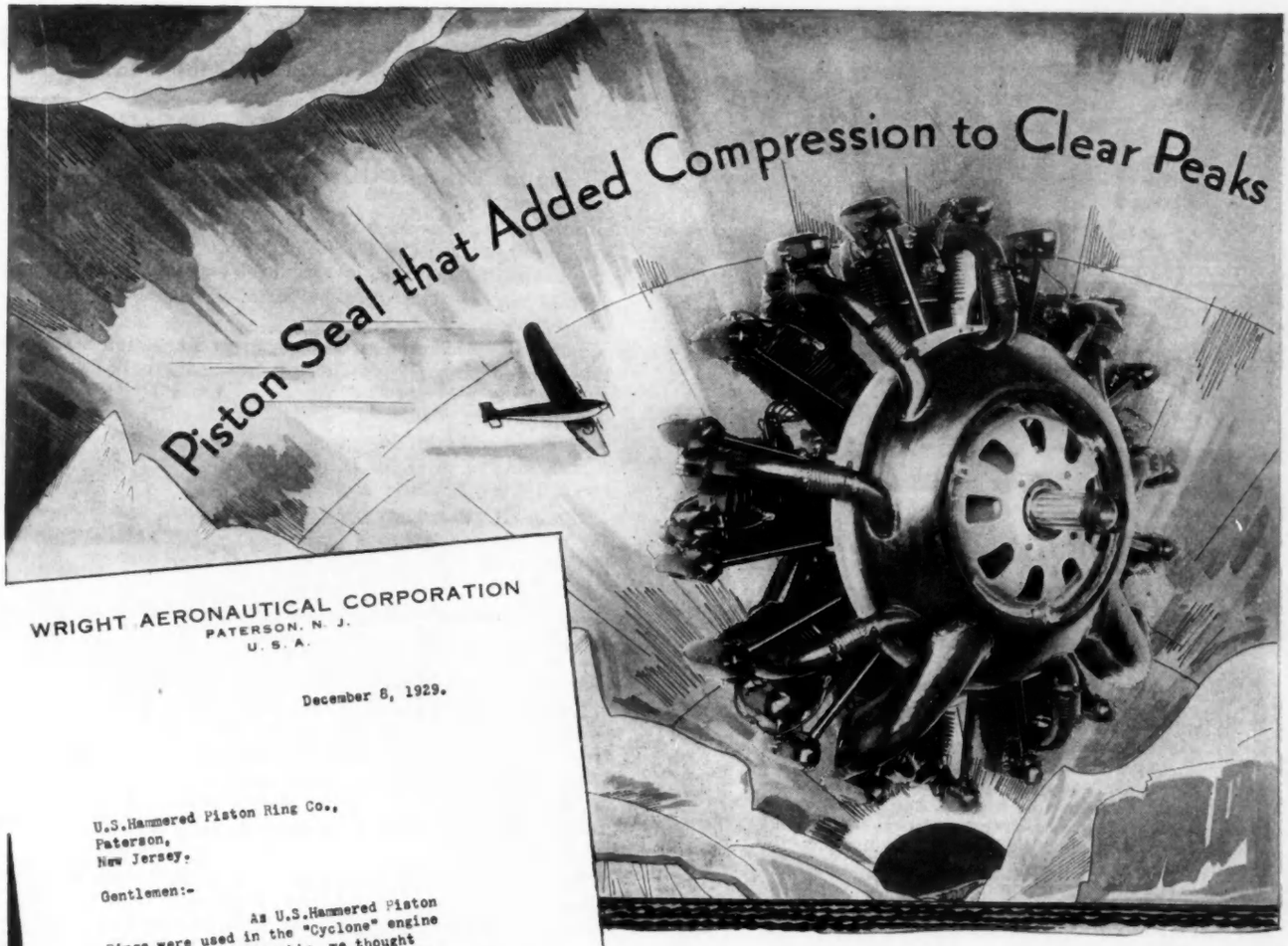
Le XXIII^e Salon de l'Automobile et du Cycle. By G. Delanghe. Published in *Le Génie Civil*, Oct. 12, p. 341, and Oct. 19, 1929, p. 380. **[L-1]**

A thoughtful analysis of French automotive conditions, both industrial and technical, introduces this account of the twenty-third Paris automobile show. The author makes a plea for the release of the French automotive industry from the burden of taxation, citing the increasing intensity of competition from abroad, especially from this Country, as the justification for such action. He calls attention to the French bureau for automotive standardization, to which the international association has entrusted the direction of international standardization in this field. Other notable developments are the organization of factory production departments, the trend toward the pooling of research activities, and the creation of chains of service stations and of specialized service stations.

Among technical developments the first to be noted is the growing importance of the six-cylinder engine, which, through the adoption of quality production, has lost its handicap of high price. The number of eight-cylinder engines is increasing in France, and the appearance of a twelve-cylinder Voisin is taken as an indication that the evolution toward an increasing number of cylinders may not have reached its final stage.

Difficulties in the fuel-feeding of the six-cylinder engine, together with the effort to endow this type of powerplant with flexibility at all speeds, has led to improvements in carburetion and fuel-feeding systems. The invasion of battery ignition, signalized at the last show, has not lost its impetus, and pressure lubrication has almost unanimous support. Crankshafts designed especially for the suppression of vibration, rubber engine-mountings, four-speed transmissions, three-speed transmissions with supplementary gearboxes, and efforts to decrease or muffle gear noise are other evidences of progress mentioned.

The descriptions of the individual exhibits are detailed and well illustrated with photographs and line drawings.



WRIGHT AERONAUTICAL CORPORATION
PATERSON, N. J.
U. S. A.

December 8, 1929.

U.S.Hammered Piston Ring Co.,
Paterson,
New Jersey.

Gentlemen:-

As U.S.Hammered Piston Rings were used in the "Cyclone" engine on Commander Byrd's ship, we thought you would be interested in a radiogram received by us from the Antarctic Expedition, a copy of which is enclosed.

Very truly yours,
WRIGHT AERONAUTICAL CORPORATION,
E.E.Hughes
Manager Sales Promotion.

EEH:MW

BYRD ANTARCTIC EXPEDITION
RADIOGRAM
RECEIVED BY
The New York Times

WFA 19 RM. Little America, Antarctica, November 30, 1929.

Mr. Roedacker,
Wright Aeronautical Corporation,
Paterson, N. J.

"Motors worked perfectly the whole trip." Bernt Balchen

"Cyclone and J5S performed excellently the whole flight" Harold J. June

Average fuel consumption 51 gallons per hour. Normal oil consumption

Particulars in my report. Regards to Mr. Kennedy, yourself and the gang.

E. J. Deans

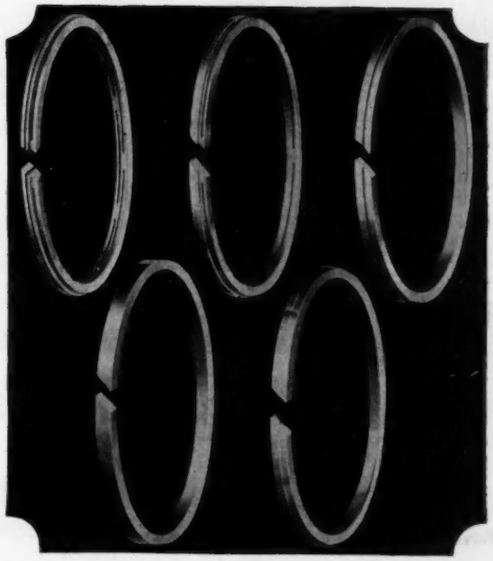
11:40 p. m.

Another Flight

attesting the supremacy of U. S. Hammered Piston Rings

U. S. Hammered Piston Ring Co.

Paterson, New Jersey



Endorsed by Leading Motor Car Builders

THE new Bishop & Babcock Motorstat has already been adopted by most of the leading manufacturers* in the automotive industry.

No other thermostat offers all these features

1. Operation from closed to fully open position through very short range, giving accurate throttling and making possible average running temperatures higher than in any usual type, with full open temperatures well below any possible boiling difficulties where alcohol mixtures are used.
2. Large bearing trunnions of special composition bronze giving unlimited wear.
3. One-piece valve design eliminating riveting and soldering.
4. Safety feature incorporated in all Bishop & Babcock Motorstats which places valve in open position in the event of a failure of the Bellows.

An experienced and complete Engineering Department is at your service ready to discuss with you design and production of Thermostatic Control for water cooling systems.

Fully covered by U.S. Patent 1696410-1644533-1590922 and other patents pending.

The Bishop & Babcock Sales Co.
CLEVELAND - OHIO - U. S. A.
LAMBERT M. PAYNE (Michigan Representative)
49 Selden Avenue, Detroit, Michigan

Names will be forwarded upon request.

Personal Notes of the Members

(*Continued from p.* 119)

Alfred Abeydeera has severed his connection as engineer with the Empire Motors, of Colombo, Ceylon, and is now serving the Rowlands Garages, Ltd., of Colombo, in a similar capacity.

Bertram L. Affleck, who until lately acted as general service manager for the Consolidated Automatic Merchandising Corp. of America, located in New York City, is now serving the Hercules Motor Corp. of the same city in a similar capacity.

Ralph F. Anderson, whose former connection was with the A. & J. Iron Works, of Chicago, where he was sales engineer, is now serving the Four Wheel Drive Auto Co., of Clintonville, Wis.

J. R. Archibald has accepted a position as wholesale representative of G. E. Jacques, Ltd., of Regina, Saskatchewan, Dodge Brothers dealers.

Elmer H. Babel was recently appointed chief engineer for the Killifer Mfg. Corp., of Los Angeles. His previous position was that of mechanical engineer with that company.

Rud J. Babor, until recently sales promotion representative for the Ethyl Gasoline Corp., of New York City, is now serving the Richfield Oil Corp., also of New York, as assistant manager of the aviation department.

Kelvin C. Backhouse, associated with the Godward Gas Generator Co., Inc., of New York City, has been advanced from draftsman and designer to assistant chief engineer.

W. Warder Bacon, Jr., has been transferred from his post as insurance solicitor for the Equitable Life Assurance Society, in Philadelphia, to the New York City office of the company.

Arthur H. Baker has severed his connection with The Trinidad Agencies, Ltd., of Port of Spain, Trinidad, B. W. I., where he was service manager, and is now machine-shop superintendent for the Apex (Trinidad) Oilfields, Ltd., at Fyzabad, Trinidad.

Charles F. Baker, formerly sales manager with the Dayton Steel Foundry Co., of Dayton, Ohio, has accepted a position with the Johns-Manville Co. in Detroit, where he is serving in the capacity of sales engineer.

J. H. Ballard, for 14 years general superintendent of the Acme Motor Truck Co., of Cadillac, Mich., has severed his connection with that company to accept a position as sales and service engineer with the Piston Ring Co., of Muskegon, Mich.

Bryant Bannister, formerly vice-president for the Bannister Motor Appliance Co. of Pittsburgh, has joined the National Tube Co., of the same city, as power engineer.

Marius J. B. Barbarou, whose connection with the Société Lorraine, of Argenteuil, France, was formerly that of technical director, has been appointed general manager of the company.

H. H. Barnes, until recently works engineer with the Durant Motors Co., in Lansing, Mich., has accepted a similar position with the Waco Aircraft Co., in Troy, Ohio.

Charles G. Barrett, associated with the Nordberg Mfg. Co., was recently transferred from the company's Milwaukee office, where he was assistant sales manager, to the New York City branch, of which he is manager.

William W. Barry, formerly sales representative of the Crowe Name Plate & Mfg. Co., of Chicago, has been appointed vice-president of that company.

Vern G. Beck, former assistant service manager of the Transport Motor Co., of Spokane, Wash., is now shop foreman of that company.

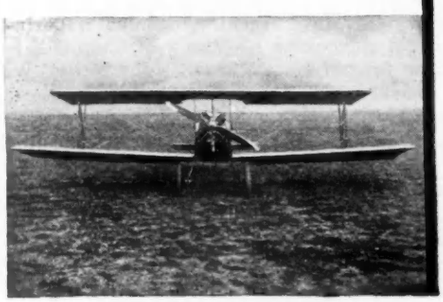
With the expansion of business of the Factory Motor Car Co., of Portland, Ore., **E. J. Blaser,** formerly proprietor of the business, has become manager.

(*Continued on next left-hand page*)

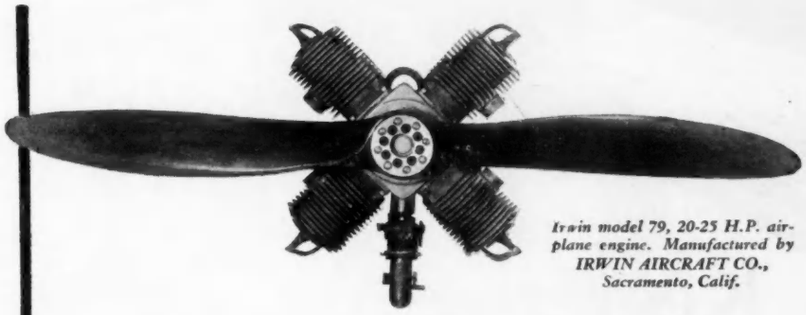
Nickel Alloy Steel Parts

used in Irwin engine:

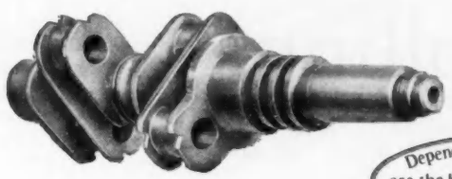
CRANKSHAFT
WRIST PINS
MAGNETO DRIVE
TACHOMETER DRIVE
VALVE GUIDES
ALL STUDS
CAP SCREWS
NUTS



Irwin Meteorplane equipped with Irwin engine weighing only 60 lbs.



Irwin model 79, 20-25 H.P. airplane engine. Manufactured by IRWIN AIRCRAFT CO., Sacramento, Calif.



Dependable materials are the best assurance of dependable performance

Crankshaft of Irwin model 79 airplane engine. Altho this crankshaft carries four throws, the entire length is only 11 inches and it weighs, finished, but 6¼ lbs. It is machined from a solid forging weighing approximately 85 lbs.

Irwin

the 60-lb. airplane engine

uses *Nickel Alloy Steel* parts

THE Irwin airplane engine, used as standard equipment in the Meteorplane, is one of the lightest weight aircraft engines now being produced in this country. It develops 20 H.P. at 1730 R.P.M. and 25 H.P. at 2200 R.P.M. and weighs complete only 60 lbs.

Nickel Alloy Steel was selected for all parts of the Irwin engine which require high strength or maximum resistance to wear. Years of industrial experience and exhaustive laboratory tests have demonstrated that the mechanical properties of Nickel Steels produced from different heats vary less than those of any other commercial steels.

In designing aeroplane engines the minimum values of the various structural materials used must be definitely known since, to save weight, section size must be pared to a minimum consistent with safety. As the mechanical properties of Nickel Steels are definitely established and are dependably uniform, their use makes possible light weight dependable design.

The widespread use of Nickel Alloy Steel parts by practically all manufacturers of airplane engines is conclusive proof of the dependability of these alloys. The experience of typical users has contributed to an extensive fund of valuable technical data. You are invited to consult our engineers and to draw upon this helpful information.

Nickel
FOR ALLOY STEEL

SEND FOR "BUYERS' GUIDE TO NICKEL ALLOY STEEL PRODUCTS"

THE INTERNATIONAL NICKEL COMPANY, INC., 67 WALL STREET, NEW YORK, N. Y.

Keep them rolling!

High speed days! In and out, through heavy traffic, the cars you sell must thread their way. Low, second, high—red light—and back to low again. And through it all the present-day automobile must give perfect, silent performance.

The bearings of the clutch release have been the special study of our bearings experts for many years. B. C. A. Bearings in the clutch release and pilot positions are an assurance of trouble-free efficient operation. That's why most American motor cars are B. C. A. equipped.

Our engineers are subject to your call.

Bearings Company of America
Lancaster, Pa.
Detroit, Mich. Office: 1012 Ford Bldg.

Personal Notes of the Members

Continued

Ernest Kaelble Bloss, former assistant electrical engineer with the Boston & Maine Railroad, in Boston, has been advanced to the post of supervisor of rail motor-cars.

F. B. Bolteuntil, recently in the Aircraft Division, Ford Motor Co., of Detroit, is now in charge of routing and materials in the planning department of the Detroit Aircraft Corp.

H. L. Brownback, who until lately was engineering director for the Brownback Motor Laboratories, Inc., of New York City, has resigned that position to perfect advanced types of aircraft engine.

William J. Bryan, who is sales manager of Heintz Mfg. Co., of Detroit, has been elected vice-president of that company.

Alfred Charles Burgoine, until recently technical superintendent for the Bristol Aeroplane Co., Ltd., of England, is now technical adviser for aeronautic engines for the E. W. Bliss Co., Inc., of Brooklyn, N. Y.

J. W. Burke, formerly garage foreman for the Sheel Co., of Long Beach, Calif., is now proprietor of the Atlas Automotive Service, also of Long Beach.

Richard L. Buysee, former motor-vehicle equipment engineer with the Michigan Bell Telephone Co., at Detroit, has been advanced to the position of motor-vehicle supervisor with that company.

Byron O. Carr, until recently supervisor of testing with Carr Brothers, automobile distributors and repairers, of Seattle, Wash., has been made a partner in the firm.

R. W. Case is now engaged as a test engineer in the physical laboratory of the Naval Aircraft Factory at Philadelphia. Previously he was aeronautic engineer of the Auburn Automobile Co., at Dayton, Ohio.

John A. Cervenka, until recently superintendent of equipment for the Hines Lumber Co., of Chicago, has relinquished his position and is now serving the Fair Co., in same capacity.

Thomas L. Cowles has been made assistant executive engineer with the Studebaker Corp., of South Bend, Ind. His post until his recent advancement was that of commercial-car engineer.

Albert A. Cripps has recently established a connection with Durant Motors, Inc., of Detroit, as a body engineer. He was previously chief body engineer with the Graham-Paige Motors, also of Detroit.

Eduardo A. Cuggia, formerly general parts service manager for General Motors of Argentina, S. A., has been advanced to the post of executive assistant to the managing director there.

Edward Peck Culver was recently advanced from his post as assistant professor of mechanical engineering at Princeton University, and is now associate professor of mechanical engineering.

Leonard R. Cutter has relinquished his position as body draftsman with the Chrysler Corp., of Detroit, and become connected with the Edward G. Budd Mfg. Co., also of Detroit, in the capacity of body-layout draftsman.

George W. DeBell, who is now an engineer with the Aviation Corp., of Buffalo, was previously aeronautic engineer with the United States Aluminum Co., also of Buffalo.

Eric W. Cornwall Densham, who is serving the Bristol Aeroplane Co., Ltd., of Bristol, England, has been advanced from the post of service manager to that of departmental manager and representative.

Roscoe L. DeSpain has given up his post as automotive equipment operation engineer with the Continental Oil Co., of Denver, to join the International Harvester Co. as district sales representative in Denver.

Frank F. Dieterichs has entered the service of the Atlas Ball Co., of Philadelphia, and is engaged on industrial methods and as experimental engineer.

(*Continued on next left-hand page*)

Mueller 600 BEARING METAL

Improves Your Product— Cuts Your Costs

In a wide range of industrial plants, Mueller 600 has solved production problems, reduced manufacturing costs and increased the efficiency of products on which it is used. Its tensile strength is from two to three times that of the best cast bronzes; it maintains a lower bearing temperature at all loads and speeds, thereby decreasing the coefficient of friction. A wide variety of tests demonstrate its superiority to phosphor bronze, cast gun metal, nickel bronze and other bearing alloys.

Some of the many purposes for which Mueller 600 has demonstrated its economy and efficiency are:

Connecting Rods, Worms and Worm Wheels, Gears and Gear Blanks, Spindles on Textile Machinery, Slip Rings for three-phase Motors, Seal Rings for Electric Refrigerators and Journal Boxes.

Mueller 600 is manufactured in the form of forgings, rod and screw machine parts. Let us show you how this remarkable alloy can promote efficiency and reduce manufacturing costs in your plant.

Mueller Brass Co.
PORT HURON - - MICHIGAN
DETROIT DISTRICT

Offices:

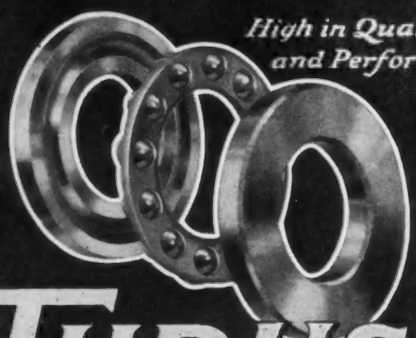
New York, Philadelphia, Pittsburgh, Buffalo, Detroit, Cleveland, Chicago, Milwaukee, Dayton, Flint, New Orleans, Minneapolis, St. Louis, Indianapolis

Mueller Brass

THREE GENERATIONS OF BRASS MAKING

AETNA

High in Quality and Performance

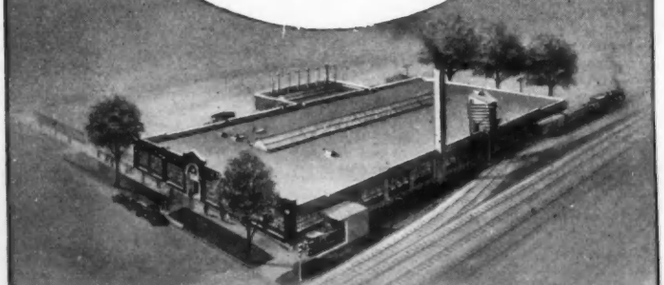


THRUST BALL BEARINGS

There is nothing romantic in the manufacture of a fine thrust bearing—to the man who buys it.

But to the Aetna organization, there is a story back of each bearing that leaves the factory—a story of skilled design and scientific manufacture.

Years of research and application to thrust bearing problems are responsible for Aetna superiority. Hence Aetna's world wide reputation as "Specialists to the Industry."



AETNA BALL BEARING MFG. CO.
4600 SCHUBERT AVE.
CHICAGO, ILLINOIS

Personal Notes of the Members

Continued

Frank G. Diffin, former president of the Langsign Corp., of Dayton, Ohio, is now president of Diffin & Co., Inc., with offices in New York City and Detroit.

S. R. Dishman, Jr., has joined the sales force of the Shell Oil Co., of San Francisco. His previous connection was with the R. L. Dunn Co., also of San Francisco, as salesman.

Lieut. Hugh C. Downey, former assistant engineering officer at France Field, in the Canal Zone, is now chief engineering officer at that field.

J. L. Dryden, former treasurer and general manager of the Long Mfg. Co., of Detroit, is now president of that company.

Ralph N. Dubois, former research engineer with the Continental Motors Corp., of Detroit, has relinquished this position to become chief experimental engineer with the Continental Aircraft Engine Co., also of Detroit.

Robert W. Duncan is now a designer and engineer for the Detroit Aircraft Corp. of Detroit. Before effecting this connection he was production manager for Gilder's, Inc., of Orion, Mich.

Leighton Dunning has accepted a position as manager of the shock-absorber division of Thompson Products, Inc., of Detroit.

R. D. Easton has been advanced from his post as machine designer with the R. R. Donnelley & Sons Co., of Chicago, and is now mechanical engineer with that company.

Earnest E. Eaton has assumed the position of research engineer for the Brown-Lipe Gear Co., of Syracuse, N. Y. He was until recently assistant chief engineer with that company.

A. W. Einstein, former assistant to Mr. Bamberger, of L. Bamberger & Co., Newark, N. J., is now statistical merchandiser for the Hahn Department Stores in New York City.

James L. Elwood has resigned as vice-president of the United Motors Service, Inc., of Detroit, and entered the parts and service department of the Chevrolet Motor Co., of the same city.

William J. Finn, now a mechanical engineer in the research department of the Texas Oil Co., in Bayonne, N. J., was previously assistant engineer of the Maxmoor Corp., of Newark, N. J.

Jack Frost has been appointed a sales engineer with the General Sales Co., shop equipment specialists, of Detroit.

Paul A. Giessel has assumed the duties of engineer with the Mahr Mfg. Co., of Minneapolis. His former connection was as inspector of mechanical equipment at the United States Engineers Depot, in St. Louis.

Edward Goodwin has entered the service of the General Motors Corp., of Detroit, as a designer, having left the independent field, in which he was a consulting engineer.

Leo W. Goosen has entered the service of Schofield, Inc., of Los Angeles, as chief draftsman. Formerly he was layout draftsman with the Miller Engine & Foundry Works, also of Los Angeles.

H. S. Graves, a former checker with the Johnson Motor Co., of Waukegan, Ill., has joined the engineering staff of the Chevrolet Motor Co., of Detroit, for which he is a contract engineer.

Ross D. Grover, former layout draftsman with the Olds Motor Works, in Lansing, Mich., has been advanced to the post of draftsman in design engineering with that company.

George R. Gwynne, formerly of Colorado, has accepted a position as transportation engineer with The White Co., of Cleveland.

Arthur S. Hawks is now acting as a consulting engineer with the Fairbanks-Morse Co., of Beloit, Wis. Before making this change he was assistant works manager of the Worthington Pump & Machinery Corp., of Buffalo.

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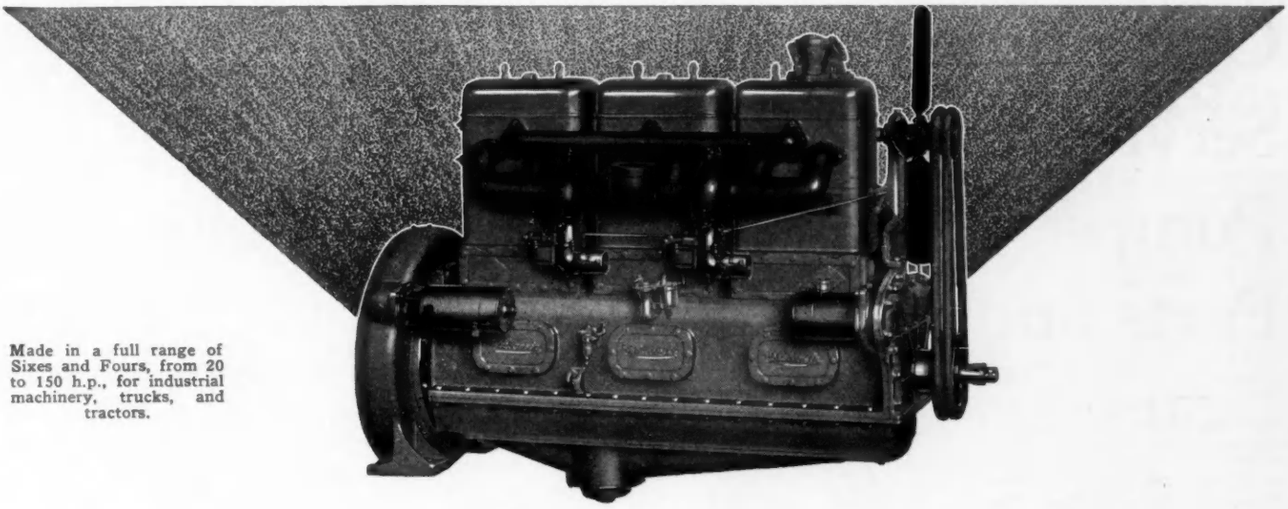


Made in a full range of Sixes and Fours, from 20 to 150 h.p., for industrial machinery, trucks, and tractors.

Steadfast Power

Unwavering steadfastness in an engine demands the utmost soundness in materials and engineering. You'll find this soundness in Wisconsin Motors. It is that inherent strength, born of over-size construction throughout, which stands up under extremes of service. Heavy going, long runs, mean operating conditions—in tropic heat or polar cold—that's when Wisconsin Motors' steadfast power is much in evidence.

Put one of these motors on your test block and find out for yourself just what it will do.

Wisconsin Motor Co.
Milwaukee, Wisconsin

Wisconsin
CONSISTENT
MORE POWER
per CUBIC INCH

Outstanding Service On Pumps Parts and Gears

The engineering and manufacturing facilities of The Logan Gear Company apply with particular force to the needs of automobile manufacturers for counsel and products that require highly specialized knowledge and methods.

Specially designed water and oil pumps.

This company is in constant volume production of oil and water pumps, special parts, and fly-wheel ring gears for automobile and engine manufacturers. The tremendous constant activity in this great industrial group is actuated only by the fact that it saves money for those who use its specialized services. We will gladly talk to you about any of your requirements for cooling and lubricating units, ring gears, or special mechanisms and parts.

Flywheel gears, ring gears of all dimensions and special gears for any purpose.

The Logan Gear Company
TOLEDO, OHIO

Logan Gears

Personal Notes of the Members
Continued

Elmer H. Heck has been made purchasing agent for the Hudson Motor Car Co., of Detroit. He was previously production car-material buyer for the Chrysler Corp., also of Detroit.

James S. Henry has relinquished his post as designer in the engineering department of the Hupp Motor Car Co., of Detroit, to become instructor in general science at Harding College, Morrilton, Ark.

Walter R. Herfurth has entered upon his new duties as purchasing agent for the Ireland Aircraft Co., Inc., of Garden City, N. Y. His previous connection was with the Freihofer Baking Co., of Philadelphia, for which he was traffic manager.

Fred W. Herlihy, former superintendent of the bus division of the United Electric Railways Co., of Providence, R. I., is now lubrication engineer with the Shell Eastern Petroleum Products Co., of Boston.

Fred C. Holz has been transferred from his post as manager of the West Philadelphia sales branch of the Gomery & Schwartz Motor Car Co., to the Harrisburg, Pa., branch of that company.

J. H. Hou, at one time special student in the factory of the Ford Motor Co., at Dearborn, Mich., has gone to Tientsin, China, where he has accepted a position as engineer in the automotive division of the Chinese Mfg. Co.

Ching Ming Hu, formerly a student at the University of Michigan, is now a structural detailer with the McClintic-Marshall Construction Co., of Philadelphia.

Victor Hughes, now located with the engineers staff of the Racine Division, Nash Motor Co., was formerly an engineer with the Johnson Motor Co., of Waukegan, Ill.

Burton R. Hurrell has resigned as general sales manager of the Willis-Jones Machinery Co., of Seattle, and is now service man with S. L. Savidge, Inc., also of Seattle.

Walter R. Jacobs, formerly engineer with the Henney Motor Co., of Freeport, Ill., has been made factory manager of that company.

Melvin F. Jackson has begun his duties as a special investigator of manufacturing and service for the J. I. Case Threshing Machine Co., in Racine, Wis. His previous connection was as special engineer for the Sundstrand Machine Tool Co., of Rockford, Ill.

Clarence E. Jeffers has resigned as chief engineer with the Martin-Parry Corp., of York, Pa., to establish offices as a consulting engineer in York.

Carl W. Johnson, former sales engineer of the Cleveland Graphite Bronze Co., of Cleveland, has been promoted to the post of factory manager.

F. M. Johnson, until recently sales engineer for the Kay Steel Wheel Co., of Philadelphia, has accepted a similar position with the Dayton Steel Foundry Co., of Dayton, Ohio.

Clarence H. Jorgensen, formerly consulting engineer with the Dole Valve Co., of Chicago, has been made chief engineer of that company.

Edward G. Juengst has relinquished his post as assistant service manager of The White Co., in Brooklyn, N. Y., and joined the Sikorsky Aviation Corp., of Stratford, Conn., where his duties are those of flight test mechanic in the experimental department.

Leon J. Jurzek, until recently manager of the lamp plant, in the airplane plant of the Ford Motor Co., at Fordson, Mich., is now superintendent with the Murray Corp., of Detroit.

Paul E. Kelekom, until lately vice-president and chief engineer of the Duvandee Vacuum Servo-Brake Co., of Liege, Belgium, is now assistant to the directors of the Minerva Motors Corp., of Antwerp, Belgium.

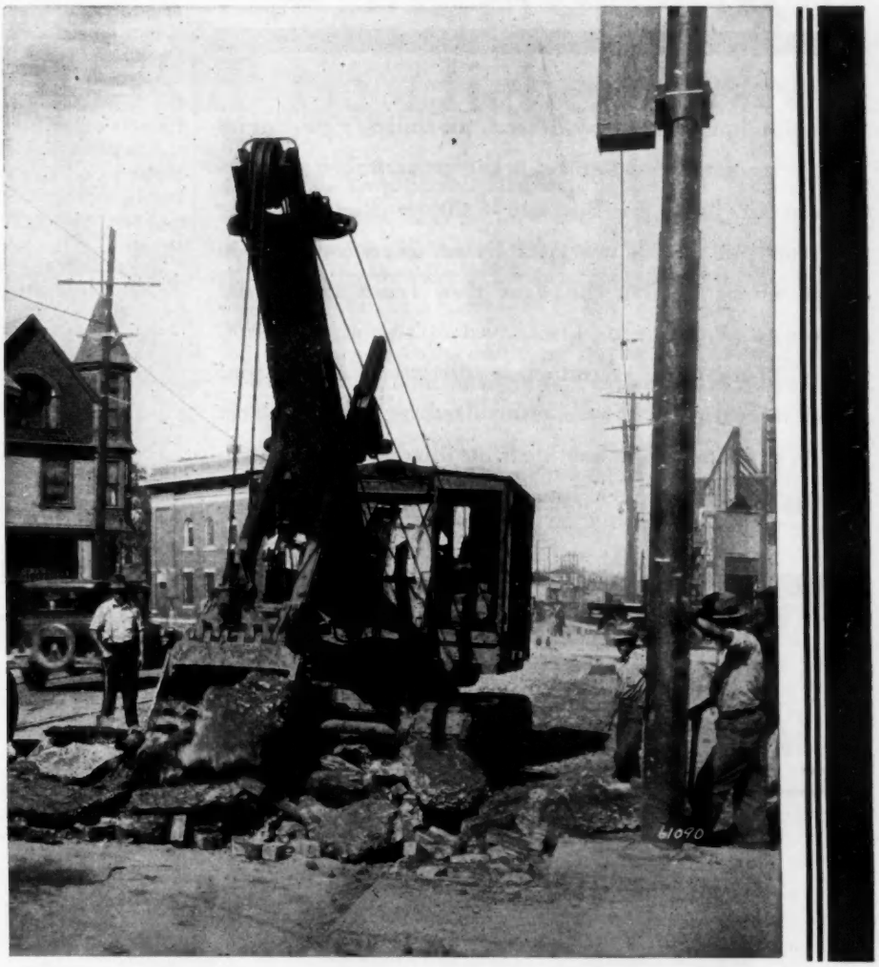
H. W. Kizer, formerly superintendent of motor-vehicle transportation with the Texas Co., of New York, is now serving the company as assistant to the manager of the sales department of the northern territory.

(*Continued on next left-hand page*)

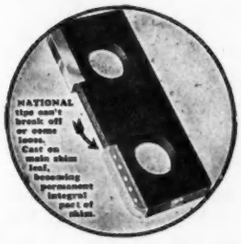
Wisconsin—an extraordinary motor

using an EXTRA-ORDINARY *Shim*

NATIONAL *Shims*



Photograph at the right shows Koehring No. 301 Shovel powered by a WISCONSIN Motor, tearing up old concrete and brick in preparation for new street paving. WISCONSIN Motors are equipped with NATIONAL *Double-Action Non-Curling* SHIMS.



With anchored Babbitt Tips (Patd.)

YOUR mistake in under-estimating the advantages of using shims is natural if you think of NATIONALS as *ordinary* shims. But they are NOT—as *Wisconsin* and 50 other leading manufacturers will tell you. They are different in *construction*, different in *application*. They reduce bearing assembly costs, increase lubricating efficiency, and speed up accurate service adjustments. Time to revise your opinion of shims. Write for complete information, and submit blueprints for prices. With or without *anchored* Babbitt Tips (Patd.) and *anchored* Babbitt Face (Patd.) For all motors.

NATIONAL
[*Double-Action ~ Non-Curling*]
SHIMS

NATIONAL
MOTOR BEARING CO., Inc., Mfrs.

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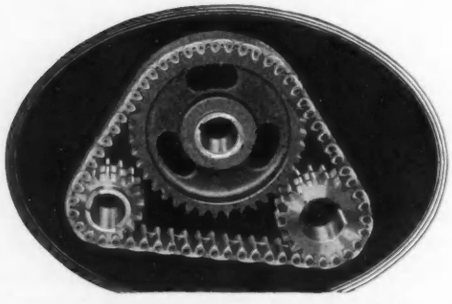
NEW YORK CITY - - 347 Madison Avenue
SAN FRANCISCO, CAL. - 460-470 Natoma St.

NATIONALS - used by more than 50 Manufacturers

THE PERMITE UNITYPE PISTON
NEW IN DESIGN
NEW IN PRINCIPLE

Dissipation of heat by design, an entirely new principle in piston mechanics, is the remarkable achievement of the new Permite Unitype Piston. The piston pin bosses are split by an ingenious cast-in slot which diverts the heat flow from the thrust faces to the extreme lower end of the piston skirt. The "four-point" contact is obviated, and initial and operating frictions minimized as never before. This great advance has been attained without weakening the piston structurally in any way!

ALUMINUM INDUSTRIES, Inc.
Manufacturers of Aluminum, Bronze and Steel Products
GENERAL OFFICES, CINCINNATI, OHIO
Detroit Office, 718 Fisher Building



SIMPLICITY

The latest type WHITNEY Silent Timing Chain is simplicity itself.

Only three different parts are required in the manufacture of this chain:

the sprocket tooth internal engaging link,

the guide link,

and the hollow shock absorbing split pin.

The patented method of link and sprocket tooth engagement in combination with the spring action of the rotating hollow split pin is responsible for the quiet, smooth running, wear-resisting action of this chain.

THE WHITNEY MFG. CO.
Hartford, Conn.

Personal Notes of the Members

Continued

Axel W. Kogstrom, who has been serving the Dunning Mfg. Co., of Detroit, as engineer, has severed that connection and is now shock-absorber engineer with Thompson Products, Inc., also of Detroit.

George A. Kraus, until recently equipment engineer for the Champion Spark Plug Co., of Toledo, has been made Detroit sales manager of the company.

Shichibei Koidzuka has severed his connection with Tokio Motors Co., Ltd., of Tokio, Japan, of which he was director and general sales manager, and is now general service manager with the Toho Automobile Co., Ltd., of Tokio, Japan.

Robert Frank Kring, a former student of the General Motors Institute of Technology, at Flint, Mich., is now a tool research engineer with the Buick Motor Co., of Flint.

George H. Kublin has given up his post of chief engineer with the Moon Motor Car Co., of St. Louis, and become an engineer with the Auburn Automobile Co., of Auburn, Ind.

Teny Kumpera, formerly general manager of J. Walter & Co., of Prague, Czechoslovakia, is now president and general manager of the Motor Cars & Aero Motors Co., Ltd., and J. Walter & Co., of Prague.

Franklin T. Kurt, former engineer with the Viking Flying Boat Co., of New Haven, Conn., was recently made chief engineer in the drafting department of that company.

John W. Lakin, formerly superintendent of maintenance for the Ternstedt Mfg. Co., of Detroit, has been advanced to the position of experimental engineer with that company.

Abbot A. Lane has resigned as production manager for the Brownback Motor Laboratories, Inc., of Pottstown, Pa., and accepted an engineering position with Stevens & Wood, of New York City.

Eric Langlands, of the Majestic Car Corp., of Cleveland, has been made general manager of the Cleveland engineering office.

Victor Larsen, former structural engineer with the New Standard Aircraft Corp., in Paterson, N. J., has obtained a similar position with the Commercial Aircraft Co. of America, Inc., at Bridgeport, Conn.

M. J. Leahy has left the independent field, in which he was known as a consultant in mechanical and chemical engineering at Fort Worth, Texas, to join the Robinson Mfg. Co., of Muncy, Pa., as a process engineer.

Yosan Lee is now a student in the Graduate School of Mechanical Engineering, of the Massachusetts Institute of Technology, at Cambridge, Mass.

LeRoy F. Le Gros, a former draftsman with the American Car & Foundry Motors Co., at Detroit, has entered the employment of the Hupp Motor Car Corp., also of Detroit, where he is a checker in the engineering department.

Otto L. Lewis, formerly supervisor of the planning department, tractor works, of the J. I. Case Threshing Machine Co., of Racine, Wis., is now chief engineer of the Progressive Motor Devices Co., of Chicago.

Robert P. Lewis, whose connection with the Salisbury Axle Co. was that of project engineer in the Jamestown, N. Y., branch, has been appointed chief engineer of the company's plant in Toledo, Ohio.

Hans W. Lindemann has joined the engineering staff of the Illinois Steel Co., at Gary, Ind., as a designing engineer.

Ivar C. Lindgren has severed his connection as service manager with the Roller Motor Co., of St. Paul, to join the Bonney Nash Motors, Inc., also of St. Paul, which he is serving in a similar capacity.

Frank J. Lucas has relinquished his position as designing and production engineer with the General Implements Co., of Racine, Wis., to accept a position with the Walker Mfg. Co., also of Racine.

Louis Lyzenga, until recently a member of the aircraft division of the Continental Motors Corp., of Detroit, is now

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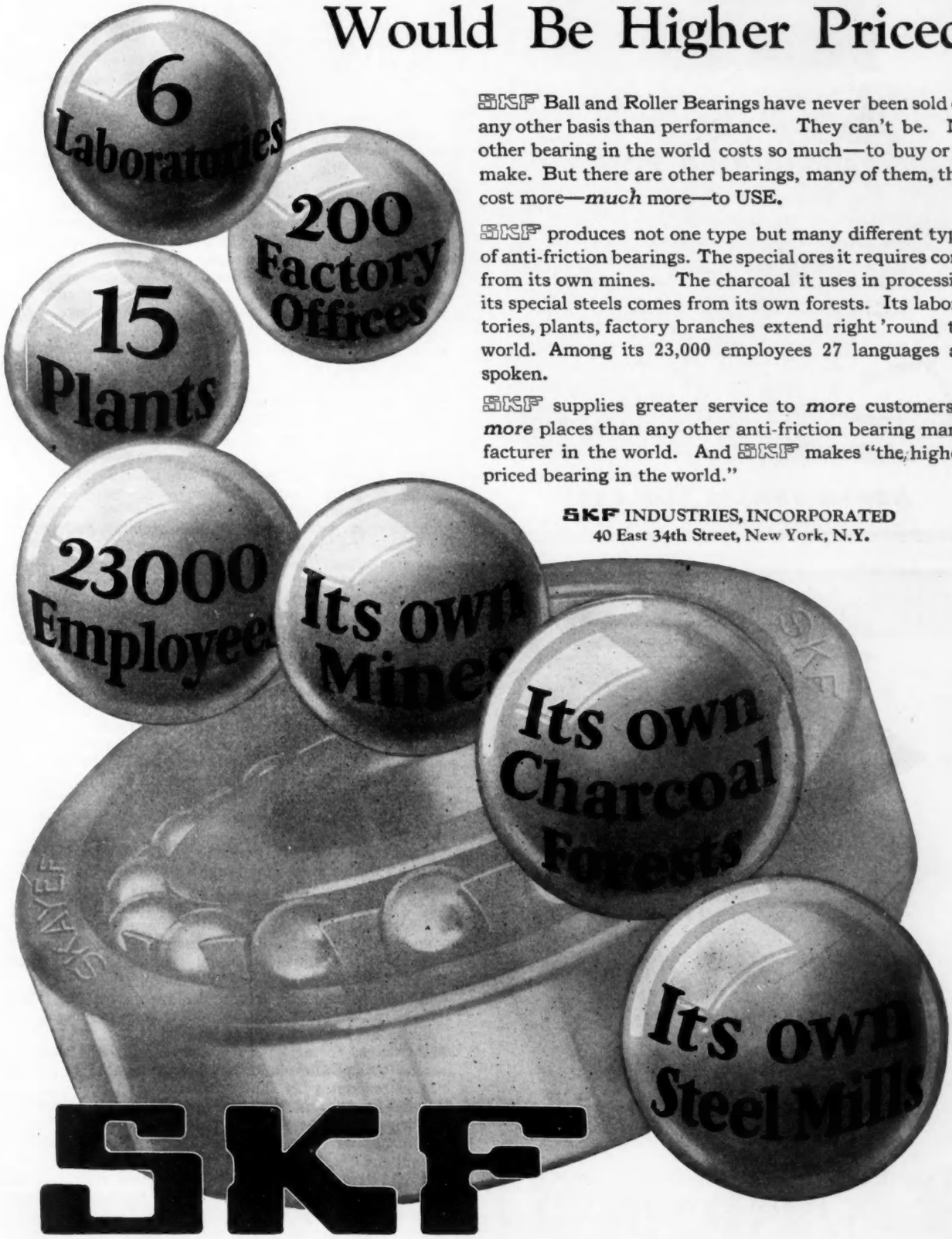
If SKF Wasn't Quite So Big....
"The Highest Priced Bearing in the World" Would Be Higher Priced



SKF Ball and Roller Bearings have never been sold on any other basis than performance. They can't be. No other bearing in the world costs so much—to buy or to make. But there are other bearings, many of them, that cost more—*much* more—to USE.

SKF produces not one type but many different types of anti-friction bearings. The special ores it requires come from its own mines. The charcoal it uses in processing its special steels comes from its own forests. Its laboratories, plants, factory branches extend right 'round the world. Among its 23,000 employees 27 languages are spoken.

SKF supplies greater service to *more* customers in *more* places than any other anti-friction bearing manufacturer in the world. And SKF makes "the highest priced bearing in the world."

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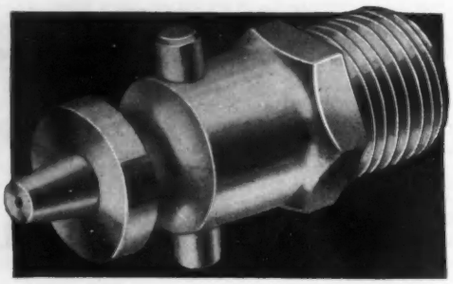
THE HIGHEST PRICED BEARING IN THE WORLD



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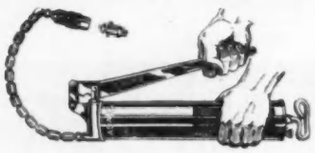
A masterpiece of scientific design—and a real boon to the car owner. Fits all types of Adams Grease Guns, including the new Push and Screw types. Widely endorsed by Engineers, Factories and Service Stations.

Samples Talk—Let us send you a few for your inspection and test. They will prove a revelation. Write for full particulars.



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Adams Grease Gun No. 11

Develops 10,000 lbs. pressure. Adaptable to all lubricating systems. Self-filling; feeds automatically. Capacity 1¼ lbs. $12

No. 14—Capacity 2 lbs. $13.50

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CRANK DROP PARK GUARANTEED FORGED SHAFTS

Counterbalanced

CRANK SHAFTS

and

Heavy Die Forgings

THE PARK DROP FORGE COMPANY

Cleveland, Ohio

Personal Notes of the Members

Continued

a member of the tool-design division of the Hudson Motor Car Co., of Detroit.

W. Stanton Marsden, automotive engineer in the sales department of the Beacon Oil Co., of New York City, was until recently automotive engineer with The Texas Co., also of New York City.

Jule F. Marshall, who has been serving the American Felt Co. as a specialist on automotive felts, has been appointed general merchandise manager of that company.

Walter Marschner has resigned his position with the Fokker Aircraft Co. of America, at Glendale, W. Va., to join Dr. W. W. Christmas in the General Development Co., of New York City, as designer and layout man for wing construction.

Ralph H. McAllister, now sales representative of the Anthracite Co., Inc., of Scranton, Pa., was until recently chief draftsman of the power department of the Philadelphia & Reading Coal & Iron Co., of Pottsville, Pa.

Stephen A. McClellan, former sales engineer with the Pratt & Whitney Aircraft Co., of Hartford, Conn., is now serving the United Aircraft & Transport Corp., of New York City.

J. J. McElroy, until recently plant manager for the Maverick Mills, of East Epston, Mass., has been made superintendent of the company.

Louis F. Merrill has recently severed his connection as chief of design in the aircraft division of the Continental Motors Corp., of Detroit, and is now chief engineer of the Axelson Aircraft Engine Co., in Los Angeles.

James Monroe has rejoined the Link-Belt Co. in the capacity of sales engineer of conveying machinery of the Cleveland district, after an absence of seven years, during which time he was vice-president and later became president of the Trailmobile Co., of Cincinnati.

Announcement has been made that **Allen W. Morton**, former vice-president of the American Hammered Piston Ring Co., of Baltimore, is now president and general manager of that company.

Glenn Muffley, consulting engineer for Copeland Products, Inc., of Mt. Clemens, Mich., has been appointed to the executive staff of American Cirrus Engines, Inc., a unit of the Allied Motor Industries, Inc., whose plant is at Marysville, Mich., where Mr. Muffley's offices are located.

James L. Myers, who until recently served the Cleveland Graphite Bronze Co., of Cleveland, as chief engineer, has been made secretary of that company.

Lorne E. A. Myles, a former production-line worker at the Fordson plant of the Ford Motor Co., is now an aircraft engine mechanic for the Canadian Wright Co., Ltd., of Montreal, Canada.

Roy Ober, draftsman, recently relinquished his position with the International Harvester Co., of Springfield, Ohio, and is now serving the Robbins & Myers Co., Inc., also in Springfield, as draftsman in the hoist division.

J. George Oetzel, until recently consulting engineer in the independent field, is now manager of the central division of the Atlas Imperial Diesel Engine Co., of Oakland, Calif.

E. E. Osterhoff, formerly a designer with the Maccar Truck Co., of Scranton, Pa., is now supervisor of the costs and stores department.

Emil Panek, formerly an engineer with the American Stamping Co., of Cleveland, has been advanced to the post of superintendent with that company.

Announcement has been made that **E. H. Parkhurst**, former vice-president and general manager of the Columbia Axle Co., of Cleveland, has been elected president of that company.

Floyd O. Paul has severed his connection with the automotive division of the J. G. Brill Co., of Philadelphia, where he was service manager, and is now district sales representative of the Carboloy Co., Inc., of New York City.

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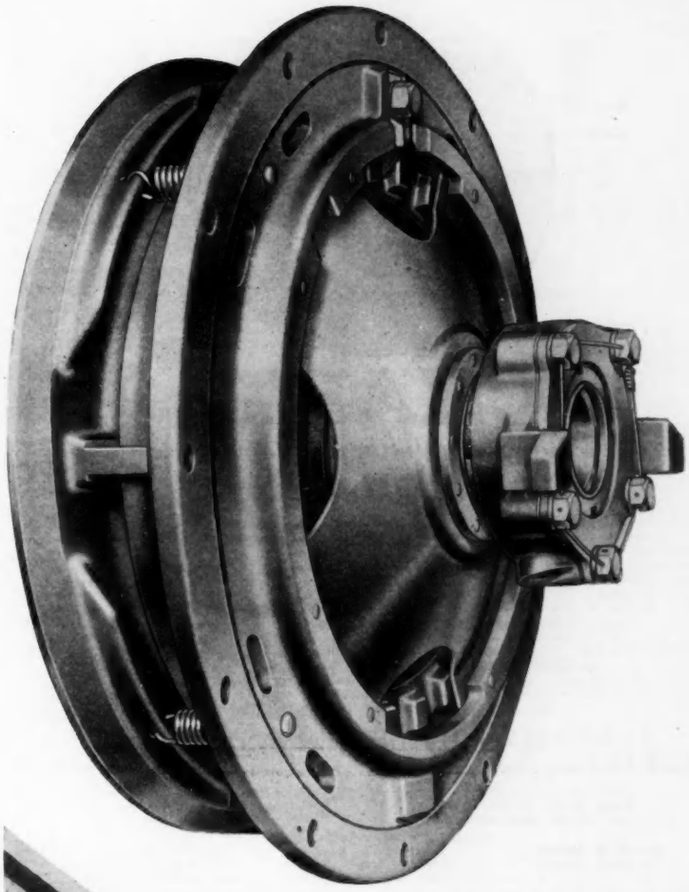


WITH competition making selling increasingly difficult, each added sales advantage is a feather in your cap. "Brown-Lipe Equipped" is a recognized advantage that will make your selling easier. Take your cars and trucks in traffic, in the open, up hills and down the straight-away. Your Brown-Lipe Clutch will be the best demonstrating feature you have.

Make the most of the selling points built into the cars and trucks you sell. Emphasize "Brown-Lipe Equipped."

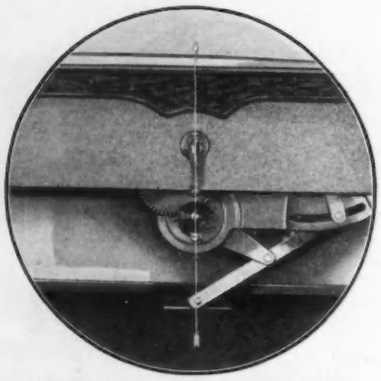
Brown-Lipe Gear Co.

Syracuse, N. Y.

Detroit **London**
San Francisco **England**

BROWN-LIPE GEAR CO.

BROWN-LIPE GEAR



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window regulator problems with
COMMON-SENSE

With characteristic reliability Common-Sense charts the solution of window regulator problems for engineers and body designers and again sets the standard for comparison.

The new Model 27 has the rapid action of the gear type, lifting the glass centrally from bottom to top, eliminating all lop-sided force and projection of corners into side channels. By using the short arm control, strain on gears is reduced one-half and sliding of arm from side to side is eliminated. Maximum rise 16⅞".

No. 6 Common-Sense Twin Arm Regulator, with its greatly increased lifting power, is ideal for convertible or partition type jobs. Numbers 5 and 10 are of the single arm type with unusual features and quality.

There is a COMMON-SENSE for any standard body. Send for our new engineering chart—it will prove there is one to meet your requirements—if not, send us your blue prints and we will design and manufacture one that will.

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ANY TYPE OR SIZE ~ ~ ANY MATERIAL

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1800 CLYBOURN AV.
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Personal Notes of the Members

Concluded

Parry H. Paul, until lately test engineer with the Philadelphia Rural Transit Co., has been advanced to the position of superintendent of the motorcoach repair shop.

G. H. Pearson, who is in this Country for a few months, has left the Australian operation office of the General Motors Export Co., of Melbourne, and is for the present at the General Motors offices in New York City.

Paul P. Pierce, until recently engineer for the Bragg-Kliesrath Corp., of Long Island City, N. Y., is now fleet engineer in the engineering division of the Vacuum Oil Co., of New York City.

Alfred W. Place, Detroit representative of the MacDonald Bros. Co., Inc., of Boston, was formerly vice-president in charge of sales and engineering for the Universal Machine Co., of Bowling Green, Ohio.

R. Spotswood Pollard, until recently New York sales manager for the Wheeler Reflector Co., of New York City, has severed his connection with that company and is now secretary, treasurer and operation manager of the Aircrafters, Inc., of Stroudsburg, Pa.

J. A. Purvis has been made technical editor of the *Automobile Trade Journal and Motor Age*, published in Philadelphia. Previous to making this new connection he was repair-shop inspector of The Texas Co., of New York City.

Frederick H. Ragan has been promoted from his post as factory manager with the Columbia Axle Co., of Cleveland, to vice-president in charge of manufacturing.

Dhiren N. Ray, formerly administrative assistant with the Leyland Motors Co., Ltd., of England, serving at the Calcutta, India, branch, has been made assistant in charge of the service department.

W. G. Retzlaff, former factory manager for the Fruehauf Trailer Co., of Detroit, has been made manager of the new transportation engineering department with that company.

William Richards has relinquished his post with the Highland Body Mfg. Co., of Cincinnati, for which he was designing engineer, and is now occupying a similar position with the American Can Co., in the same city.

Marcus P. Robbins is now assistant division manager in the power department of the Monongahela West Penn Public Service Co., having been advanced from his post as assistant to the president's assistant of that company.

G. H. Robinson, who is serving the Durant Motor Co. of Michigan, has been transferred from the Detroit branch to the Lansing branch.

Walter W. Rogers has recently become a sales engineer for the Johns-Manville Corp., of Chicago. His previous connection was as sales engineer and Michigan representative of the Multibestos Co., in Detroit.

Thomas B. Roscoe, until lately service manager for the Parvin Motor Co., of Warren, Pa., is now service manager for the Charles I. Rice Co., of Hackensack, N. J.

Albert E. Ruff, who was until recently brake research engineer for the American Chain Co., of Bridgeport, Conn., is now a draftsman in the powerplant section of the General Motors Corp. Research Laboratories, at Detroit.

Marvin H. Rutherford, a former student of the University of Wisconsin, is now employed as assistant mechanical engineer with the Chicago Transformer Corp., of Chicago.

Maurice Sainturat, until lately chief engineer for the Donet Zedel Automobile Co., of Neuilly, Seine, France, is now engineer with Chenard & Walcker, of Gennevilliers, Seine, France.

Otto R. Schoenrock, now director of engineering for the Oliver Farm Equipment Co., of Chicago, was formerly chief engineer of the O. R. S. Engineering Co., of that city.

Waldimir Semenyna, until recently connected with the Gates Day Aircraft Corp., of Paterson, N. J., as a designer, has formed a new connection with the New Standard Aircraft Corp., of Paterson, as project engineer.

Charles M. Serre, who has been serving the Société des Usines Renault, at Billancourt, Seine, France, as technical director, is now chief engineer with that company.